BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[621.138.5 (.43)]

Note on the preparation of locomotive repair programmes,

by Dr.-Ing. Fr. LANDSBERG, Reichsbahnoberrat, Berlin.

In proportion as industrial undertakings extend and as their administrative and technical services become more highly specialised, the subdivision of the work is more and more inevitable. At the same time, this gives rise to the risk of the dispersion of energy, to internal friction, and to indecision in the management. Consequently Henry Fayol, the pioneer worker in administrative philosophy, was quite correct in formulating one of his rules for directing industrial businesses as follows : « a single head and a single plan for whole series of operations having the same objective in view [1].

In the case of railways, the various departments of which are important owing to their extent and the work to be done, this rule ought to be observed with particular exactitude. The primacy of the Operating Department, which sells transport and exercises therefore a direct action on the receipts is, it is true, recognised without discussion. However, it is the duty of the other departments to act in such a way that the traffic is handled under the best possible conditions from both the technical and financial points of view. Centralisation of the issue of orders, in the higher sense attributed by Fayol to this expression, demands above all the exact knowledge of the relations between the different departments and the linking together of their functions. This knowledge is necessary, not only to ensure the undertaking being managed economically, but also to draw up a correct programme (budget).

As an example, we propose to show for the important departments « User » and « Maintenance » of the rolling stock, in other terms for the technical services of the locomotive running and workshops department, the importance of determining the relations just mentioned.

These two spheres of action require to be studied with particular care, because the stock of vehicles and the workshops represent a large capital investment and because it is essential, for proper management and for economical operation,

(1) Fayol, Administration industrielle et générale (*Industrial and General Management*), 1916.

that the orders be distributed by a common agreement. There is such close contact between the two services that the knowledge and consideration of the two points of view and the complete harmony of the effort of all those concerned (managerial and executive) towards the common end should be required, as being indispensable. This is the more difficult as the service (integral use of the vehicles) responsible for meeting traffic needs, in the spirit of the remarks made above upon the primary importance of the traffic department, also claims for itself the leading position.

Within the limits of these general observations, we will consider one case only, that of the locomotives; the methods described can be applied, however, directly to the other rolling stock, so long as the conditions regarding shopping it are the same.

*
* *

General repairs to locomotives, that is to say heavy repairs prescribed by the railway company's own regulations, or by official regulations, require the vehicles to be taken out of traffic for a considerable period and form the basis of the activity of the workshops. The periodicity of such repairs is therefore of capital importance as regards the regular and economical working of the locomotive running and maintenance departments. The necessity of replacing the locomotives during the time they are under general repairs has its influence on the amount of rolling stock required. Furthermore, as the locomotives must be repaired within a rather short period of time, which has to be followed uniformly, they affect the output of the shops (equipment and number of workmen). The consequence is that a programme is essential for both services.

This programme is governed by the regulations dealing with the kind and period of the repairs and should adapt itself within these limits to the different circumstances which can arise and to the organisation rules of the service.

The preparatory work can be carried out in two ways :

a) Owing to the diversity of the influences at work, it is more likely to be done accurately if these factors are arrived at individually, that is to say by compilation and comparison of the rostered working of the different vehicles. In this way, the local character of the service, according to the operating and permanent way conditions, is taken into account at the same time and shown in the form of a relation between the tractive effort, the distance run, and the period of service;

b) In the case of areas, the configuration of which can be expressed numerically or which are large enough for the different influences to cancel out themselves, the preparatory investigation can be carried out by means of *average values obtained from the statistical returns*, always provided these contain the necessary data. We get the closer to reality as the subdivision by kinds of use (classes of trains) and classes of locomotives is conceived and carried out with greater care.

In both cases, when considering the programmes, not only must the locomotives in actual operation, but also those in running order and ready for service be taken into account. Three questions must be considered in this investigation :

the number of vehicles,
the class of work,
the working condition.

The *class of work* is always shown — even when a wide material subdivision by class of work is not possible — by the statistics showing the result of the use

thereof, that is to say not only the mileage run in unit time (month, quarter, etc.) but also the mileage run between two general overhauls or, in other words, in the maintenance period.

The *working condition* is intended to show the state of the vehicle at any date up to the maximum mileage fixed by the regulations (in Germany for example for passenger vehicles) or the average mileage between two general repairs (for locomotives), this value being determined from the statistics. It is expressed in the form of a fraction of the maximum value; for example a 75 % locomotive has already run 75 % of the average distance laid down for the group or class.

The connection of the three above mentioned characteristics between themselves and with the stock of locomotives in service and in reserve will be considered further on. As the relations are not easy to appreciate, graphs are used to the greatest possible extent and we have started from elementary theories and ideas. Obviously the diversity peculiar to railway operation, and which has to be allowed to continue without restriction if the necessary elasticity is to be safeguarded in unexpected cases, cannot be considered under all aspects in a general study of this kind. On the other hand, it must not be forgotten that the different partial phases are governed by laws which must be known if we are to get optimum results, both as regards being constantly ready for service and for economical working.

I. — Principles of the investigation.

Figure 1 shows in the system of distance-time co-ordinates, the ideas and methods of representation we will use throughout the article. The locomotive, new or thoroughly repaired, is worked a certain distance first of all to run it in,

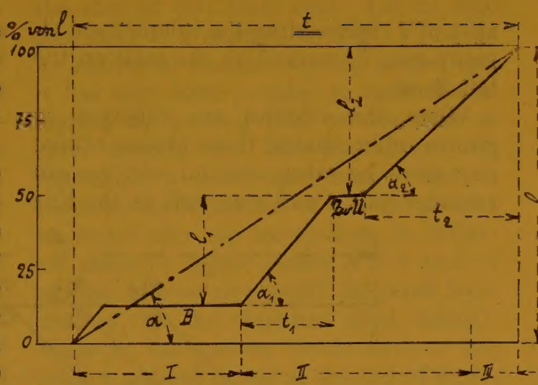

Fig. 1. — Distance run by locomotives between two general repairs.

I. — Initial period.
II. — Active period.
III. — Final period.

and is then stored as a spare engine. When definitely put into service, it is run until the intermediate repair with a factor of utilisation expressed in terms of time by $\tan \alpha_1 = \frac{l}{t_1}$ and after the intermediate repair with a smaller user, $\tan \alpha_2 = \frac{l}{t_2}$; these hypotheses are used to make the example a general one. The engine reaches the average distance in kilometres laid down in the regulations given by the statistics after l kilometres in a period of time t (expressed in months for example). The statistical values l and t are ascertained for the principal classes (express, ordinary passenger and goods trains locomotives, subdivided again between tender and tank engines).

Until it has run a distance of 0.1 l, the locomotive is considered as being *new* or as *spare* (initial period); after more than 0.9 l it is considered as *run down* (final period) and between these two periods as *used* (active period). The working condition of the vehicles, that is to say the proportional part (as a percent-

age) of l representing the kilometrage already run, is marked on the scale on the left hand.

When the vehicles are collected in groups or in classes, these phases cannot any more be taken account of : we can consider only l and t as well as the line connecting the beginning and the end of the maintenance period as shown in the general statistics.

In this way, for a stock of locomotives used under the same conditions or for parts of such a stock, we get the diagram, figure 2, on which the locomotives newly

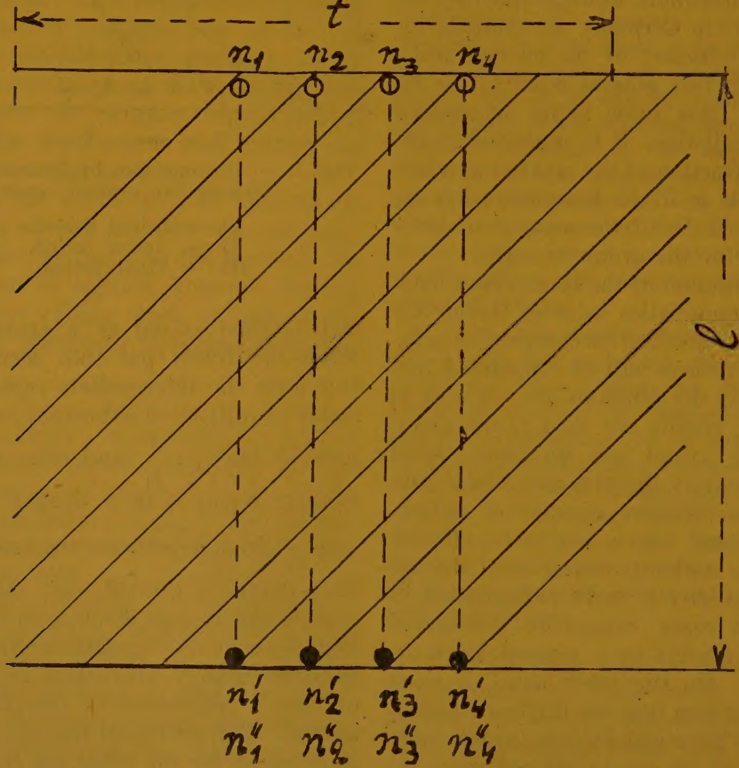

Fig. 2. — n_1 to n_4, general repairs due.
n'_1 to n'_4, locomotives returned to service after general repairs.
n''_1 to n''_4, locomotives newly put into service.

put into service at equal intervals (month, quarter, etc.) are shown by groups with their kilometrage between two heavy repairs. The different groups include n_1, n_2, n_3, etc., locomotives and the engines in each group come in for general repairs all together after a kilometrage l and at the end of a period of time t. The group n_1, going into shops, is usually replaced by n'_1 locomotives; it undergoes general repairs in proportion to requirements and, when these are finished, the group is again at the disposal of the operating department. For greater clearness, the period out of traffic is not shown on the diagrams given below,

seeing that, in accordance with the fundamental considerations before us, we can presume it is covered in a suitable manner. Obviously, this must be taken into account in practice.

At each moment we find as total of the groups met with by a vertical intersection the number of locomotives in service and at the same time for each group the working condition at that time. The limit case of possible regularity occurs when $n_1 = n'_1$, $n_2 = n'_2$, etc.; in the case represented by the figure, the number of locomotives is then $Z = 8\ n$.

We can most easily study the irregularities always occurring if we start from the idea of this limit case. Let us suppose that at a moment *a* (figs. 3 and 4) n_1 additional locomotives are needed and it has only been possible to meet the situation by putting into service *new* or *spare* locomotives. This hypothesis also represents a limit case, seeing that to avoid an accumulation of general repairs we could also use locomotives in different states of repair unless it is a case of putting into service newly delivered locomotives (cf. the diagrams given below). These n_1 locomotives would have to undergo a general repair at the end of a period *t*, in *b*, and would at intervals overload the maintenance department. Below we

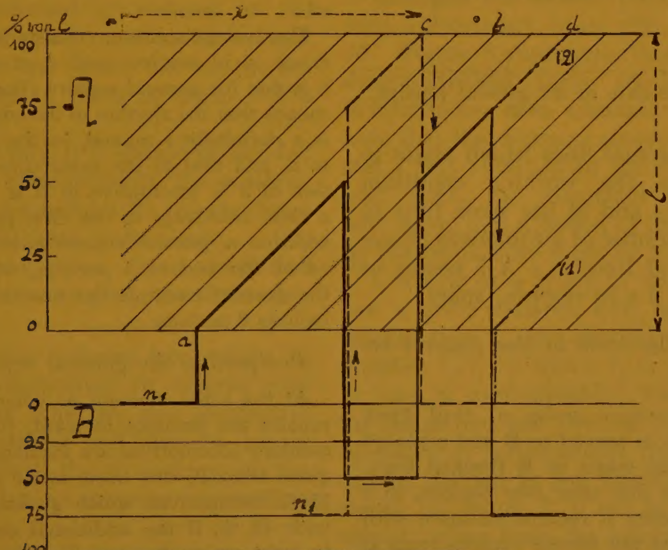

Fig. 3. — Advancing the date of the general repairs.

consider the additional n_1 locomotives shown on the figure by a second line and we propose to leave over the supplementary additional repairs to another time. This step can become necessary when at the moment *b* the operating department would not be able to adapt the working to suit the deficit resulting from engines waiting general repairs, or to the shops only being able to carry out the general repairs with some delay. For adjustments to the made, an exchange must be made between groups in different states of repair and this can only be done by *using reserve engines.* These latter engines are shown on the lower part of the

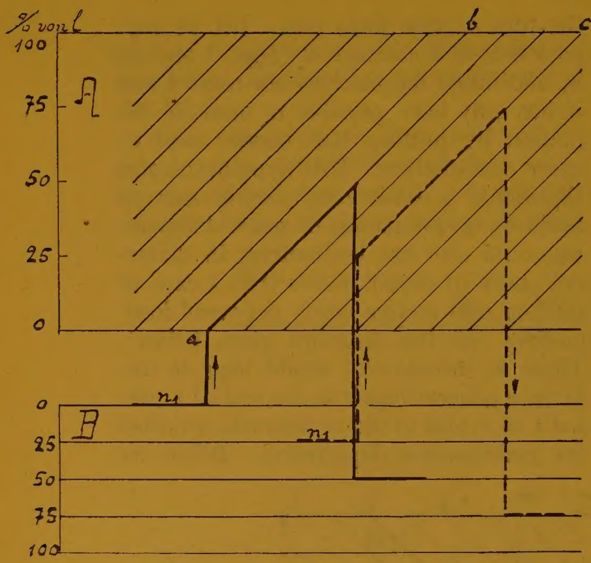

Fig. 4. — Retarding the date of the general repairs.

figure as horizontal lines which ought to correspond to the working condition (vertical scale) and at the same time to indicate the number of locomotives (number or width of lines). « A » means in service, and « B » in reserve (spare).

Advancing the time of the general repair (fig. 3).

The supplementary group n_1 (full thick line) with 50 % is moved to B and replaced by a 75 % group taken on B (broken line). As the latter is due after two periods, in c for general repairs, it returns as spare with 0 % in B, whilst the former returns from B to A up to the moment *b*. If at this moment the additional demand ceases, the supplementary group is shown at B with 75 %.

The following is the result of these operations.

The general repairs of n_1 locomotives are brought forward from *b* to *c*.

From starting the exchange until *b*, B is made younger.

In particular, n_1 spare locomotives (with 0 %) are already available starting from *c*, whereas they would have been available from *b* if no exchange had been made.

At the end of the cycle, there are again as at the beginning n_1 locomotives with 0 %, and n_1 locomotives with 75 % at B, these two groups having meanwhile changed over their working condition.

If the supplementary demands persist beyond the moment *b*, we have to choose between two solutions :

We revert to the same position as at *a*, by moving the group n_1 with 75 % to B, and by replacing it by the group with 0 %, which is then put into service (line 1).

As a result of this, the movement from *b* to *c* of n_1 general repairs is again made good because it is followod by a proportionally longer period and the operation is not repeated.

The second solution consists in leaving the group n_1 in service until *d*, at which period it is due for general repairs (line 2), and this means that the rhythm of the additional group is « definitely » moved, in the time, from *b* to *d*, and that at the same time the proportion of 0 % locomotive in B is increased for a time relatively to the first procedure. In addition n_1 general repairs have been effected at the end of 4 periods, whereas under the present example the normal time would include 8 periods.

Postponing the general repair (fig. 4):

At the same moment as when the general repairs are brought forward, the n_1 supplementary locomotives are exchanged with engines from B, but these latter are this time 25 % locomotives which go back at *b* to B with 75 %, if the additional demand ceases to exist, or continue until *c*, for general repairs, if the additional demand continues.

The general repair can be put off a second time by means of B, by replacing for example at the moment *b* the n_1 75 % locomotives by 50 % locomotives.

The result of these measures is the following :

In the event of a temporary additional demand, the general repairs of *n* locomotives are not carried out, as B meets the additional kilometrage ; consequently when the supple-

mentary need ceases to be felt, B possesses n_1 50 % locomotives and as many 75 % locomotives, instead of n_1 0 % locomotives and as many 50 % locomotives at the beginning.

If the additional demand continues the general repairs are again put off to the extent that B can provide for the additional kilometrage.

The exchange between locomotives in service and locomotives in reserve can also be justified at a given moment — independently of the date at which the general repair ought to be undertaken — by questions of an economic nature affecting the locomotive running department; such is the case, for example, when the loads are being increased or reduced or the timings altered, and more suitable types of locomotives are taken out of the reserve and used in place of those which are in service.

In the same way, if it should be decided to rejuvenate systematically the working condition of the engines in reserve, some exchange might have to be made.

We can draw, from these simplified though typical examples, the following conclusions :

1. Permutation between locomotives in service and locomotives in reserve provides a means whereby the locomotive running department and also the repair shops can adjust their working to the economic conditions they are required to meet.

2. In order that such permutation be economical, the working condition of the whole of the locomotives must be known and taken into account.

3. When general repairs are carried out earlier than usual, the condition of the locomotives in reserve is improved, whereas when these repairs are postponed the condition of the spare engines deteriorates.

4. By such permutation, the number of general repairs can be reduced or increased during certain periods; this may occur once or — to meet seasonal needs — at regular intervals.

5. The stock of engines in reserve is decided by the number and working condition of the locomotives included in it.

These principles have been deduced from the relations existing between the groups of locomotives which have run the same mileage in unit time (l/t) and the reserve stock. However, the reciprocal relations of the groups of which the l/t's differ among themselves, under the hypothesis that the existence of the two groups is justified for service reasons and reasons of economy, must also be considered. Figure 5 represents this case in the diagrammatical form employed above. Whilst it has been admitted tacitly up to now that the distance covered in kilometres l was obtained in the limit of time laid down, the important condition is now added that a period of time t_z figured by the regulations must be utilised.

A vehicle or a group of vehicles having run a great distance l_1/t_1 in wich t_1 is smaller than t_z, and another with a small distance l_2/t_2 in which t_2 *is* greater than t_z, may be exchangeable. The limit case, the simplified but unusual one in which, by means of permutations, the two groups can be carried to the usuable time t_z is shown on figure 5; it can be understood straight away from the information given on the figure.

There is no alteration in the period of overhaul, but there is an exchange of the locomotives to be repaired at the dates originally fixed. The above mentioned introduction of the reserve engines into this cycle of exchanges makes the locomotive working a very flexible one, and one adaptable to the circumstances, besides making it possible to take into account the most widely differing condi-

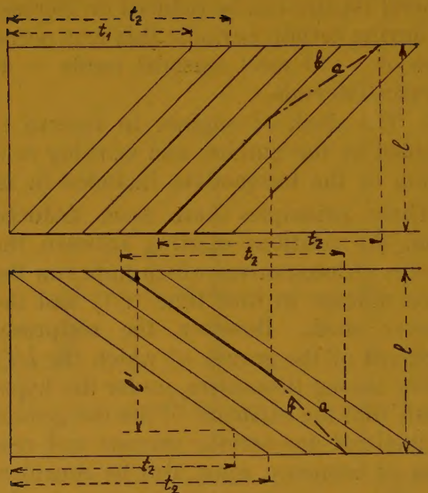

Fig. 5. — t_1 t_2, working time expected.
$t_{\bar{2}}$ usable working time.
l' usable kilometrage without exchange.

tions, such as the alteration of the general repair periods, the use of the locomotives in given although variable workings and the full use of the working periods, the prompt putting into service of spare stock, etc.

II. — Application of the principles to the statistics.

When it is a question of following from a statistical point of view and of studying in advance the periods the general repairs become due for wide areas or for the whole undertaking, it is not possible to consider the different individual phases which can be likened to « molecular movements in the organisation ». It is preferable to look for points of view which will make it possible to appreciate their consequences as seen outwardly so as to be able to study the previsions made and verify them. What we said above shows clearly that we must know the working condition of the reserve stock — at least — the average condition — to be able to understand the whole position.

We will consider separately the previsions and the verifications thereof. We must first of all bring into evidence the limits of the accuracy of the statistical figures.

Limits of accuracy.

The kilometrage l and the time t between two general repairs are determined at the last general repairs by the workshops department and recorded in the returns. Owing to the difference in the way a given locomotive is employed in working different kinds of trains, the data can no longer be grouped and used for statistical purposes according to the work on which employed, but only by the types of the locomotives. As a rule, we can limit ourselves to the principal classes (express, ordinary passenger train locomotives and goods train locomotives, subdivided as between tender and tank engines). In the estimates which naturally are based on the expected traffic and which, therefore, relate to the kilometrages grouped by classes of trains, it is necessary to use *a posteriori* conclusions on the classes of locomotives needed for this purpose, or to group the classes of trains so as to be able to compare the said main classes of locomotives without too great error. We need hardly say that there can be no question of this approximation if we can ascertain the possibility of using, and the effective use, of the different locomotives, the local conditions being taken into account (lines, services, timetables, etc.).

The use of the principal classes of locomotives in the different classes of trains is revealed in the average statistical figures supplied for them of the values l and t.

The principal classes have therefore a

ratio l/t, statistically equal, and we can retain for them, considered as statistical groups, the method of representation so far applied; the number of locomotives can also be replaced by their kilometrage.

Checking.

The examination subsequently carried out is intended to take into account the alterations in the locomotive stock, at the same time linking them up with the distances run in service and with the repairs, and to bring out *grosso modo* the way the programme has been followed or departed from (especially also as regards sending the engines into the workshops for general repairs).

The adaptation of the programme, possible in details and made use of as need be by the local services, to the variable demands of the Operating Department which as we have already shown, has many needs, can also be followed by examining its effects.

If in the limit case of great regularity the time interval between two general repairs is divided, for example as shown in figure 2, into 8 periods and if the total number of the locomotives is constant and equal to Z, at the end of each period a group of $n = \frac{Z}{8}$ locomotives must be taken into the shops for general repairs : as each locomotive brings with it l kilometres, the contingent due for repairs, expressed in kilometres, is $n \times l$. In this particular case, the calculation can also be made direct as follows : in the preceding time t, the 8 groups together have run $8\ n\ (l/t)\ t$ kilometres, which for a period or for a group represents $(n.\ l)$ kilometres.

We must not forget, when studying this question, that for the locomotives in a group the average statistical figure l/t is a resultant as shown in figure 6.

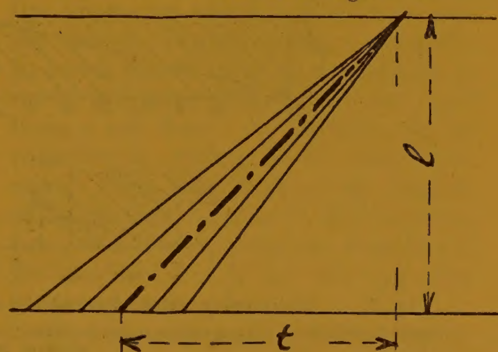

Fig. 6. — Computation of the average statistical value.

If the different groups do not include equal numbers of locomotives, or what amounts to the same thing, if the number of kilometres (for the same ratio l/t) is not divided equally between the m periods of the time elapsed t, it is not easy to appreciate at first sight how the batch due for repairs is to be determined. Applying the argument just used, we will look for this batch by dividing, by the number m, the total kilometrage run by the locomotives in the time t. We obtain in this way, for any given moment, the batch to be repaired by means of the folowing equation :

$$\frac{n_1 \cdot \frac{l}{t} \cdot t + n_2 \cdot \frac{l}{t} \cdot t + n_3 \cdot \frac{l}{t} \cdot t + \text{etc...}}{m} = \left(\frac{1}{m}\right) \Sigma_n. \qquad (1)$$

First of all we will see, using figures 3 and 4, if we have obtained suitable values for checking purposes, whilst in particular taking account of the interference with regularity of working resulting from the exchange with the reserves (changes in the dates of general repairs).

In this example, a group has been

strengthened by n_1, relatively to the number n of locomotives in the other groups : in other terms the number of kilometres run by this group has increased by $n_1 . l/t$ during the time t. This hypothesis holds good naturally in the case of alteration of the time of general repairs. At a certain moment, therefore, the $n_1 . l$ additional kilometres must be made good by means of heavy repairs.

If using equation 1, we calculate the kilometres to be expected to need general repairs at the moment considered, the influence of the additional kilometrage on the batch to be repaired begins to be felt at the point a. In figure 7 we have only

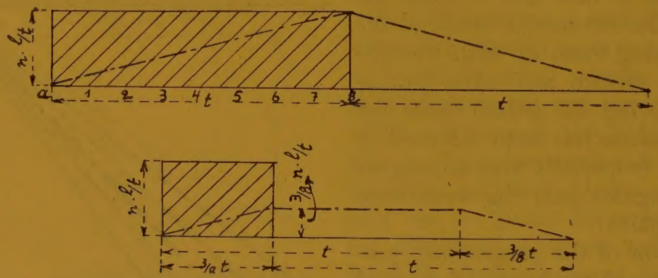

Fig. 7. — Preliminary calculation of the date on which general repairs come due.
Hatched surface = excess kilometrage. — Mixed line (dots and dashes) = calculated additional general repairs.

shown the kilometrage in excess (in the upper part at a, for example, on figures 3 and 4) using as ordinate a period namely $n_1 . l/t$. The total number of kilometres to be expected is consequently, at point 1:

$$\left(8n \cdot \frac{l}{t}\right)\frac{1}{8} + \left(n_1 \cdot \frac{l}{t}\right)\frac{1}{8} = \frac{l}{t} \cdot \left(n + \frac{n_1}{8}\right)$$

at point 2 :

$$\frac{l}{t}\left(n + \frac{2n_1}{8}\right)$$

and so on.

At point 8, the ordinate of the additional batch must have become great enough to be in line with $n_1 . l/t$: it will fall in the following time t, according to the same law, to 0 (mixed line).

The area of the triangle having for base $2 . t$ is :

$$\frac{n_1 \cdot l}{t} \cdot 2t \cdot \frac{1}{2} = n_1 \cdot l$$

and is consequently equal to the total additional kilometrage.

In the lower part of the figure (at b) we show how an additional kilometrage of shorter duration ($3/8\ t$) in this case, influences the calculation of the batch due for repair. The ordinates increase up to $3/8\ n_1 . l/t$, remain constant during the time (t — $3/8\ t$), and then fall from $3/8\ t$ to 0.

The area is in this case :

$$\left(\frac{3}{8} \cdot n_1 \frac{l}{t}\right)\left[\frac{2}{2} \cdot \frac{3}{8}t + \frac{5}{8}t\right] = \frac{3}{8} \cdot n_1 \cdot l$$

i. e. it again equals the total additional kilometrage.

We see that the additional distance run is reflected in the calculation of the batch for repairs during the time of this additional kilometrage and the subsequent time t and that as a result this distance is extended in time. As in the general consideration of statistics, as has been pointed out already, the alterations in the dates of general repairs are not known

and may therefore have been made prior to or after the additional kilometrage, this calculation with the graphical representation, covering a greater period of time than that of the supplementary distance, is of great value. The areas bounded by the ordinates representing kilometres per unit of time, and by the abscissæ showing the time, correspond — as we have shown — to the total distance run and are therefore directly comparable with the areas in which the kilometres recovered by general repairs are represented by ordinates at the end of the different periods. This enables us to ascertain at any moment how far the general repairs have been anticipated or put off by the intersection of the areas of batches ready for general repairs and of those having undergone general repairs. It is of the very nature of the use of the locomotives that for a given moment the exact agreement of the two batches is the exception : the comparison must be over periods of some length.

For the practical application of the process it must be pointed out that the locomotives taken out of service whether finally (by scrapping) or temporarily (to be stabled as being unserviceable) should be considered as run down and withdrawn with the statistics 1 of the number of kilometres calculated for the batch to be repaired. For 1, in this check the effective statistical value of the time elapsed must naturally be introduced. This being done, the comparison of the areas « to do » and « done » becomes easy; the differences are explicable solely by the locomotive kilometres added or borrowed from the reserve. The kilometres « repaired » in excess should result in a reduction in age or in a numerical reinforcement, and those « not done » in increased age, or reduction in numbers of the stock in reserve. These considerations are completed by the recapitulation of the kilometres run. When comparing the situation at the periods I and II the following must be taken into account :

The kilometres used up of the B (spare) locomotives at the moment 1 + kilometres due for general repairs of the locomotives in service between I and II (net, that is to say after deductiong eliminated locomotives) = kilometres used up of the reserve locomotives at the time II + kilometres run since the general repairs during I to II.

The mean working condition of the reserve is given by the ratio :

$$\frac{\text{Kilometres utilised of the B (spare) locomotives}}{\text{Number of B (spare) locomotives.}}$$

Comparisons of this kind ought, as figure 7 would seem to indicate, to cover periods of not less than 2 t. In an area of some size there are, however, so many peak distances of different kinds due to local circumstances that, by balancing these peaks, the above considerations can be applied to shorter periods : this is, moreover, essential in order to avoid surprises.

As l and t change, as the construction of the locomotives, the organisation of the locomotive running department and the organisation of the repair shops are improved (l grows for t either constant or slightly falling) from time to time and at least once a year, these values must be calculated from the statistics and the corrected values on which to base the different periods be used.

The method described above suffices, therefore, to solve the problem set at the

beginning, which consisted in following up continuously, from the statistics, the kilometrage run and the general repairs from the elapsed, and to draw therefrom the necessary data on which to base the steps to be taken.

Preparation of the programmes.

Measures based on probable traffic developments can only be adopted in the maintenance department when taking as the starting point the results of the check. The ratio of the batches due for general repair and those generally repaired during a certain time already elapsed as well as the working condition of the reserve stock of locomotives are the determining factors.

If, for example, the general repairs are in arrears, and the spare engines are getting old in date, the equilibrium must be re-established by advancing the general repair by using the method of exchange indicated above; this measure ought only to be put off to a later date if a reduction in distance is expected, but even then the check is essential.

By making use of the reserve, the peaks of short duration can be met although the general repairs will have to be made good subsequently. The estimation of the value of the reserve then depends upon its composition as upon the magnitude and duration of the peaks, which have to be met. The following consideration

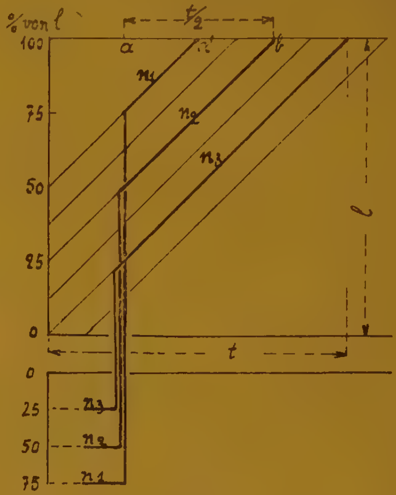

Fig. 8. — Composition and value of the expected kilometrage of the reserve stock.

shows this clearly (fig. 8). Let us suppose that we have in a group which is expected to run a certain distance in the unit of time l/t :

n_1 (75 %) with an available service time of 1/4 t.
n_2 (25 %) » » » 2/4 t.
n_3 (25 %) » » » 3/4 t.

The sum of the available kilometres is then :

$$(n_1 . 1/4\, t + n_2 . 2/4\, t + n_3 . 3/4\, t)\; l/t$$

This total cannot, however, be directly related to the kilometres to be expected as additional kilometrage.

If, for example, the nature (start and height) of the peak requires all the locomotives to be used suddenly ($n_1+n_2+n_3$), the n_1 locomotives can only be used for ¼ t, and then must undergo general repairs, that is to say they must be replaced by others.

When the peak lasts $t/2$ the reserve is used as shown in figure 8. The gap (a—b) in the general repairs of the n_1 locomotives cannot be made good either, though part of the total number of kilometres remains in

excess; this cannot be used because it only becomes available after the peak has passed (after b).

Before making a proposed increase, for a limited time, in the kilometrage of the locomotives, we must begin by calculating the number of locomotives needed to meet the situation. If the number of locomotives in reserve with a service availability equal or greater than the duration of the peak is not sufficient, with the necessary margin of safety, the reserve locomotives which have worked a longer period must go into the works in good time for general repairs, so as to be ready for service at the beginning of the peak period (that is to say, in the case of figure 8, the n_1 75 % locomotives).

If, on the other hand, the reserve shows an excess, it is possible as we have shown above, to fix in good time beforehand with a view to assisting the repair shops, the dates the general repairs are due to the locomotives already having done a fairly long service.

If the kilometres run are to be increased for a prolonged period, the number of locomotives in reserve must particularly be considered. In the following example we can admit that, the time of service t being divided up into 8 periods (quarters), the number of locomotives in service in each of these periods is increased by x and that the reserve includes :

$2\,x = n_1$ 75 % locomotives, that is with a still available period of service of $1/4\ t$,

$3\,x = n_2$ 50 % locomotives, *i. e.* with $1/2\ t$, and

$1\,x = n_3$ 25 % locomotives, *i. e.* with $3/4\ t$.

As before, no allowance has been made in the drawing for the time taken in carrying out general repairs. The progressive absorption of the available kilometrage in the reserve engines is shown in figure 9.

With the supposed composition of the reserve and the succession, shown in the drawing, of the putting of the locomotives into service, the additional demand can be satisfied theoretically, in the limit case for the time $5/4\ t$, by the reserve itself, if this demand does not exceed $6\ x$ and then remains constant (see top part of the figure).

The maintenance service has to accept, through the general repairs to the locomotives coming out of reserve, an additional load which, starting from 2, is uniformly x general repairs per quarter, with interruptions at 3, 8, 9 and 11. These interruptions must be taken advantage of to generally repair the locomotives coming in no longer fit for working, and which now must be used as supplementary. By judiciously working in this reinforcement and by making exchanges, we can naturally continue for a longer time to meet the operating department's needs, which is not indicated in the drawing. On the other hand, this latter, considered as a theoretical limit case, enables us to appreciate the aggravation of the situation by the fact that, starting from 5, the locomotives recently given general repairs must be put into service straight away (since the reserve stock is in full use on the line): a peak in the transmission of the stock due for repairs only occurs at 13 with $2\ x$.

We have started with $6\ x$ locomotives in reserve, the available working time of kilometrage of which was about 46 %, that is : $1\ (x.\,3/4 + 3\ x.\,2/4 + 2\ x.\,1/4)$: $6\ x$ in %.

As an instructive comparison, we will now examine the case in which, to obtain the same result, we start from the hypothesis that the reserve possesses the same

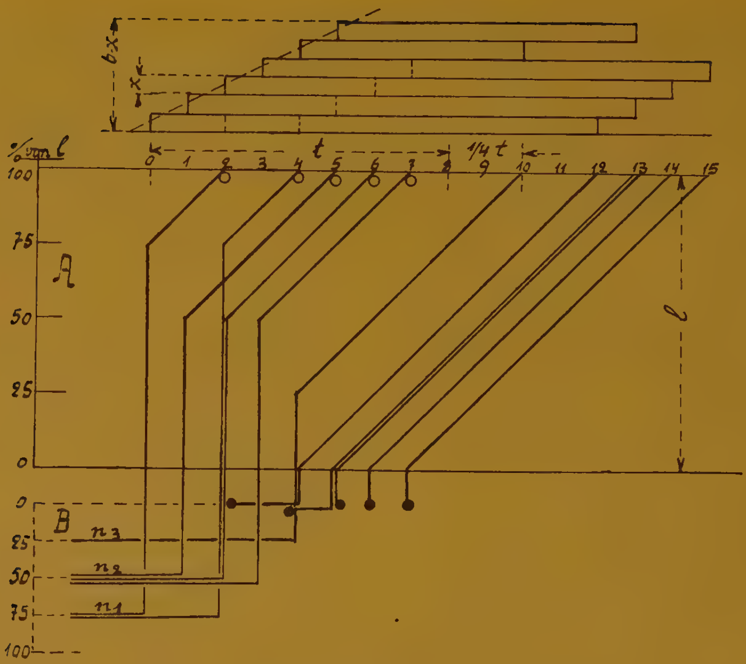

Fig. 9. — Limit kilometrage of the reserve.

Reserve $\begin{cases} n_1 = 2\ x \text{ with } 1/4\ l \text{ remaining,} \\ n_2 = 3\ x \text{ with } 2/4\ l \text{ remaining,} \\ n_3 = 1\ x \text{ with } 3/4\ l \text{ remaining.} \end{cases}$

number of available kilometres, but this time in locomotives which have done no work (0 %).

We must then have $11/4\ x = 2\ 3/4\ x$ locomotives having available l kilometres. The lower part of figure 10 shows that by the 3rd quarter we shall be obliged to complete the stock with 0.25 x locomotives taken from the reserve, and that in each of the 4th, 5th and 6th quarters we shall be obliged to draw therefrom with the same object, x locomotives, whereas starting from point 8 we can use those coming back from general repairs in replacement of those going into the shops for general repairs. Consequently, to meet the same additional demand as above (6 x and 5/4 t), 6 x locomotives are also required. Relatively to the first example, the kilometres available at the end of 5/4 t is slightly greater (19/8 $x.l$ instead of 17/8 $x.l$); the same remark applies to the number of general repairs now become necessary (5 1/4 instead of 5).

These examples prove the fact, shown by practical experience, that for an equal number of available kilometres, a greater number of spare locomotives, more or less worn, is more useful as regards quick adaptation to the requirements and shows more flexibility than a smaller number of locomotives which have done no work. Independently of this, it is still desirable, as we have pointed out several times, to send to the repair shops in good time before an expected additional demand, the run-down locomotives and give them general repairs so as to increase the kilome-

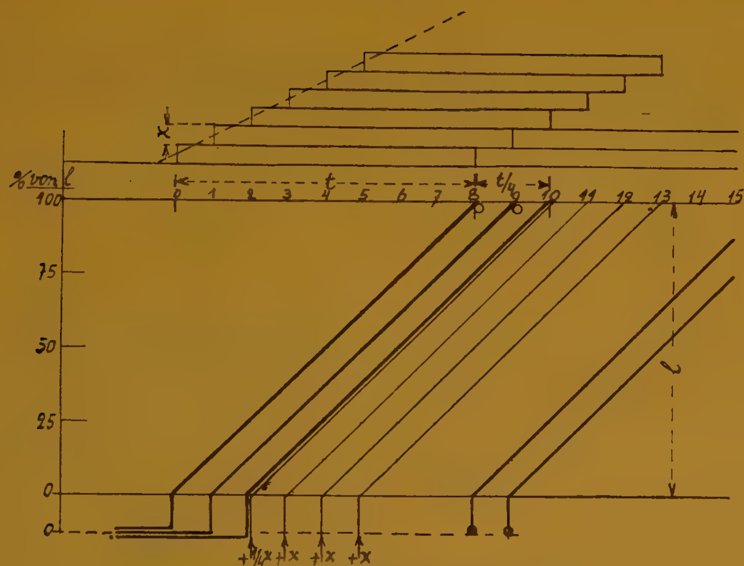

Fig. 10. — Limit kilometrage of the reserve.
Reserve $n = 2\ 3/4\ x$ with l.

trage available in the spare engines, and to put the spare stock into such a state as to be able to meet sudden demands, which generally cannot be foretold with complete certainty.

When drawing up a scheme or programme, a prudent estimate of the improvements through improved methods in the values l and t will be taken as basis; the use of the actual values obtained in the previous period, that is to say as a general rule l remaining smaller and t greater, offers the best guarantees for certainty.

Summary.

In the above considerations we have shown, for two neighbouring spheres of action and in relationship thereto, their natural interconnection which must be known if the investigation of the problem of the *use and the maintenance of the vehicles is to be carried out in common.* These studies are used as the basis for the *common allocation of the orders:* they even constitute in themselves the ultimate sequence of orders and determine the arrangements to be made for the departments concerned under all headings (economic organisation of the staff, of the rolling stock, finances, repair work in the shops, etc.). At the same time frequent checks are required at close intervals to ensure that the programmes retain sufficient flexibility to meet unexpected modifications of the theories on which the study has been based.

[621 .131 .2]

Ratios of modern locomotives, (*)

by A. I. LIPETZ,

Consulting Engineer, American Locomotive Company.

(From the *Railway Mechanical Engineer.*)

The type E superheater, the long travel valve motion and certain changes and refinements in proportions have tended to increase the power of modern locomotives beyond that indicated by Cole's ratios. The author develops a revised method of easily applied coefficients for use with modern locomotives.

* * *

No one method or formula can express the tractive force of all locomotives. So many variables affect the performance of a steam locomotive that attempts to devise a universal formula have never been successful. As an example, it suffices to say that in designing steam cylinders the proper dimensioning and shaping of steam ports and bridges, and the relative location of exhaust- and live-steam passages, are of great consequence as regards the question of losses caused by wire drawing and condensation. No formula can express the ability of the designer to provide an economical cylinder. The most that can be done by a formula is to express the average results of a well-proportioned locomotive of a design that is characteristic of a certain period of locomotive development, and to give an idea of the power of a steam locomotive under ordinary working conditions with a degree of accuracy sufficient for practical purposes.

With this end in view, the author attempts in what follows to provide means of quickly and fairly accurately evaluating the horsepower and tractive force of a modern two-cylinder, simple-expansion locomotive equipped with a superheater of sufficient size, with or without a feedwater heater, with a proper valve motion, and with other parts characteristic of modern locomotive design.

For locomotives of an earlier period, Cole's formulas are sufficiently accurate, and although the method about to be explained could also be extended to cover all locomotives, no attempt has been made to supersede the Cole method for locomotives of pre-war design.

Tractive force.

The basis of all tonnage and speed-time-distance calculations is the tractive force of a steam locomotive. Of the different tractive forces that are sometimes being used, the basic one is the « indicated tractive force, » which is the direct result of the work of steam in the cylinders. The « rail tractive force, » which is the imaginery force applied to the rim of the wheels, and the drawbar pull on the tender drawbar, can be found from the indicated tractive force, if the engine friction and total locomotive resistance are known. The object of this paper is to establish a method and figures for the calculation of indicated tractive force as a function of speed.

* Abstract of a paper « Horsepower and Tractive Effort of Steam Locomotives (Locomotive Ratios) » presented at the 1932 annual meeting of the American Society of Mechanical Engineers at New York. Contributed by the railroad Division.

A method of plotting the tractive-force curve on the basis of cylinder sizes only, using the rated tractive force and certain factors, irrespective of the size of boiler, is in principle incorrect. Nevertheless, in practice, it can give good results for the reason that in a locomotive of a certain period and of conventional proportions, the sizes of the locomotive boiler and cylinders are in a certain ratio to each other. However, as with improvement in locomotive design the boiler becomes larger in proportion to the engine, in order to meet the increased speed requirements of traffic, the formulas based on cylinder sizes and Cole's factors begin to give too low values.

Mr. Cole himself introduced the idea of boiler horsepower. He determined it to be equal to the maximum evaporation in pounds per hour divided by 20.8 for superheated steam, this figure representing the consumption of steam in pounds per horsepower-hour, including auxiliaries (1). This made it possible to determine what is called the boiler percentage, which means the ratio of maximum cylinder horsepower (at 1 000 ft. per min. piston speed) to the boiler horsepower, the maximum cylinder horsepower being found from the Cole formula, and the boiler horsepower as stated above. Cole recommended designing locomotives with boiler percentages as close to 100 per cent as possible. In this case his method of plotting the tractive-force curve would be correct, strictly speaking, at least for one point, namely, 1 000 ft. per min. piston speed. However, for locomotives with larger or smaller boilers, the boiler percentage had no effect on plotting the curves. Therefore some locomotive builders and railroads have already found it necessary to make corrections in the Cole factors in relation to the boiler percentages. This in fact is a roundabout way of figuring the correct tractive force for a piston speed of 1 000 ft. per min., and an approximate value for other speeds.

Direct methods of establishing tractive-force values.

There are at least two possible direct methods for establishing the tractive-effort values for different speeds —one, which might be called the analytical method, and the other, an empirical method.

The first, the analytical method, is logical and seemingly very simple; however, in order to give good results, it requires a number of corrections, which can be determined only from actual experience. It would seem very simple, if the evaporation of the boiler and principal dimensions of the locomotive engine are known, to calculate the amount of steam per stroke for each speed and the tractive force for that speed resulting from the work of the figured amount of steam.

This method was proposed and used in the early years of locomotive development, and for a long time it was considered to be the only possible method in view of its comprehensibility. It is rational in principle, and therefore had a great many adherents. The difficulty with the method, however, lies in the fact that two important and very complex factors are neglected : first, the cooling of steam in the cylinder, and second, the distortion of indicator cards due to speed, both of which cause considerable losses.

The cooling of steam in the cylinder, or what is known as the heat exchange between the steam and the cylinder walls (and heads), cannot be expressed by any theoretical formula. A large number of studies have been made, some of them

(1) *Locomotive Ratios*, p. 9; also *Locomotive Handbook*, American Locomotive Co., 1917, pp. 64-65.

employing very complex forms of mathematical analysis, in order to establish the cooling effect of the cylinder, but none of them has given practically satisfactory results, as too many variables are involved. With highly superheated steam the conditions are slightly better and some formulas can be established, but even then, resort must be had to empirical figures from tests, as will be shown later.

The second factor, distortion of indicator cards, depends upon the valve-motion characteristics, steam-port dimensions, and steam passages, as all of them entail drops in pressure, depending upon the speed. These laws are also very intricate and do not easily lend themselves to mathematical expression; and when formulas are used, they must also be corrected by test data, which of course are only approximate.

The second method, which is an outcome of just this consideration, was suggested also years ago by locomotive investigators. The advantage of the empirical method, expressed either by tables or formulas, lies in the idea of basing the calculations on horsepower rather than on tractive force, as the former depends upon a smaller number of variables than the latter. If the locomotive is built in accordance with certain standards of perfection and refinement, the horsepower depends mainly upon the amount of steam generated by the boiler. The sizes of cylinders do not come directly into consideration. If they vary within a limited range, as is the case in locomotives, and if the proportions are correct, they should not influence the steam consumption per H.-P.-hour. As is known from the theory of steam engines, the important factors are the cut-off and the number of revolutions per minute. The first factor determines the expansion ratio and the thermodynamic efficiency of the steam cycle, while the second affects the condensation and steam-friction losses.

Some investigators think that steam consumption depends on piston speed, and so Mr. Cole found in his investigation of locomotive data, but the majority of steam-engine authorities refer steam losses to the rotary speed of the crank and not to piston speed. Steam-consumption figures based on recent locomotive test data also show more uniformity when they are referred to crank speed rather than to piston speed.

However the case may be, the advantages of referring the steam consumption to piston speed, or to the r.p.m., cannot be great, as the piston strokes do not differ much in modern locomotives, being between 28 inches and 32 inches. On the other hand, as the cut-off depends on the r.p.m. and not the piston speed, it seems to be more practicable to consider the steam rate as depending on the r.p.m.

The tractive force of a locomotive is one of the two components of the locomotive horsepower. The cylinders and driving wheels permit resolving the horsepower into tractive force and speed in accordance with

$$P = T \times V/375 \quad \ldots\ldots [5]\,(^2)$$

Thus the tractive-force calculations can be simplified by determining first the locomotive horsepower as a function of either the r.p.m. or the piston speed, depending upon which of the two influences on the efficiency of the locomotive is reflected in the steam consumption per H.P.-hr., and then later resolving the horsepower into tractive force and locomotive speed.

The foregoing, of course, refers to the boiler tractive-force curve, which has a hyperbolic shape. As to the horizontal portion of the tractive-force curve, this is determined entirely by the rated trac-

(2) Although some of the formulas have been omitted, the author's numbering has been retained. (Editor, ***Railway Mechanical Engineer.***)

tive force. There is no connection between these two except only as far as the sizes of the cylinders and boiler in a well-proportioned locomotive are dependent upon each other.

Boiler tractive force.

Thus the tractive-force curve depends in its major part upon the amount of steam generated in the boiler and the consumption of steam per horsepower-hour. The natural thing, therefore, is to establish the laws of the two variables. The analytical method can hardly lead us to the result sought for, and we have to rely entirely upon test figures.

Unfortunately, tests are not being conducted in such a way as to enable us to establish the two above-mentioned variables as completely as is desired. Even the most elaborate tests, which are supposed to give all data in which an investigator should be interested, attempt to find solutions for various problems not always with a view to establishing the tractive-force curve. However, by analyzing available data, certain conclusions can be drawn, as will be seen below.

Strictly speaking, horsepower and tractive force are not definite conceptions unless we specify certain conditions at which power and tractive force are being considered. There can be the *maximum* tractive force which the engine is capable of producing irrespective of the amount of coal burned and the cost of such a forced operation. There can be the *most economical* tractive force, and this from the point of view of fuel only, when short cut-offs and long expansions are used, or from the point of view of total operation expenses. The latter is the more important, and it is safe to assume that under ordinary conditions the mode of operation of locomotives, established by long experience, is in the end the most economical. This can be called « performance tractive effort ». Only in exceptional cases, when the locomotive is pulling a fast passenger train at top speeds and is forced to the limit of its power, is the maximum tractive force, or what is called the « capacity tractive force, » developed.

The method of plotting the maximum and average performance curves is clear from figure 15. Both horsepower and tractive-force test figures have been used and mutually checked by formula [5].

It would seem that stationary tests should give the figures of maximum evaporation and speed, because each test is conducted at constant cut-off and constant speed, and the most severe combination of these two variables should be close to the limit of boiler capacity. With this in view, curves were plotted for five Pennsylvania locomotives (K4s, E6s, K2sa, L1s and I1s), as shown in figure 1, for different numbers of revolutions of the driving wheels per minute. The data were taken from bulletins published by the Pennsylvania Railroad on locomotive tests at the Altoona testing plant. For better comparison, evaporation is given not in absolute figures but in percentages of evaporation for respective boilers in accordance with Cole constants (American Locomotive Co. Hand-book, 1917, p. 59).

It will be seen that while the curves differ very widely, their shapes all have the same parabolic characteristic of increase with speed up to a certain limit, and subsequent drop for higher speeds. The peak points are from 12 to 28 % above the Cole figure. This must have been due to forcing the boilers to high evaporation rates. The peak point on curve 1 (locomotive K4s), for instance, corresponds to a rate of firing of 171 lb. of coal per sq. foot of grate area per hour. On curve 5 (locomotive I1s) the rate of firing is still higher, due to the small grate area of this locomotive. In figure 1 the rates of firing for peak points are inscribed in rectangles. The

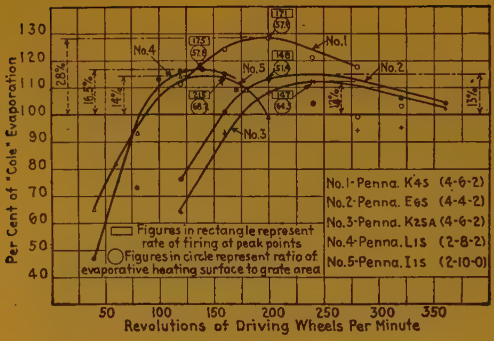

Fig. 1.

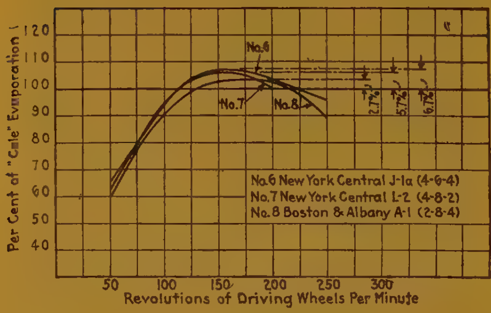

Fig. 2.

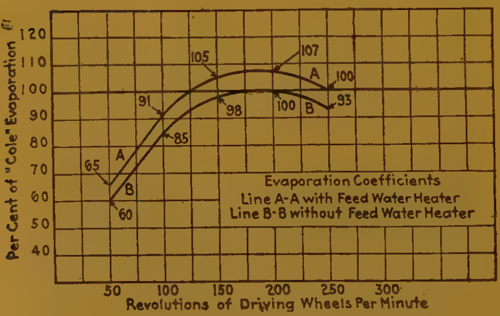

Fig. 3.

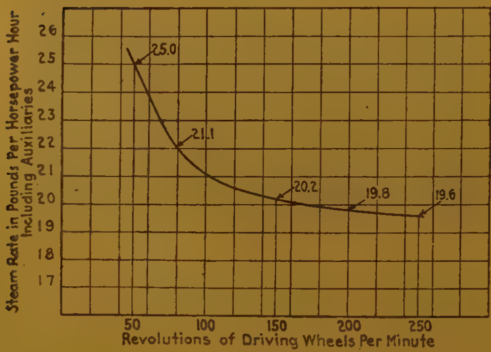

Fig. 4.

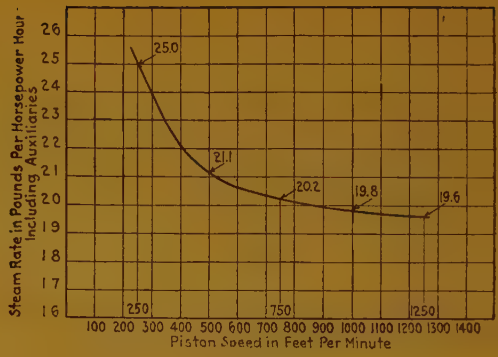

Fig. 5.

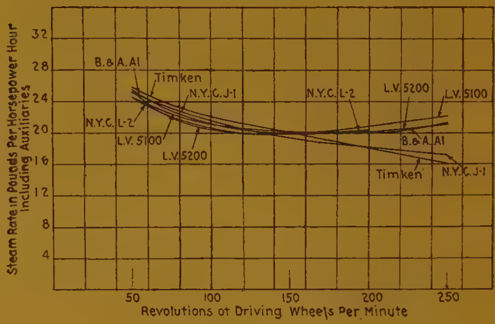

Fig. 6.

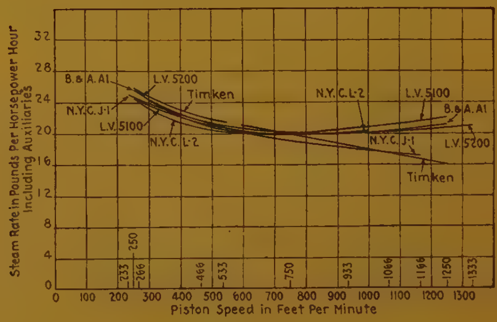

Fig. 7.

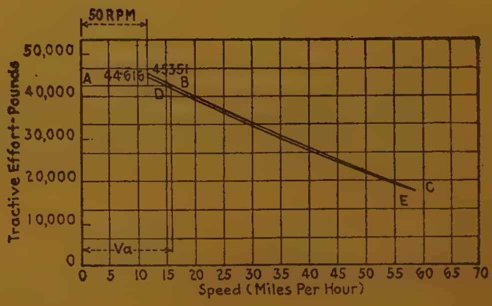

Fig. 8.

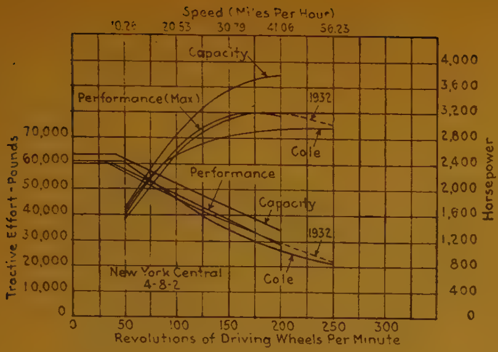

Fig. 9.

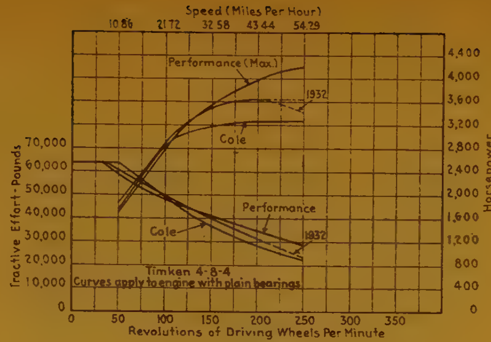

Fig. 12.

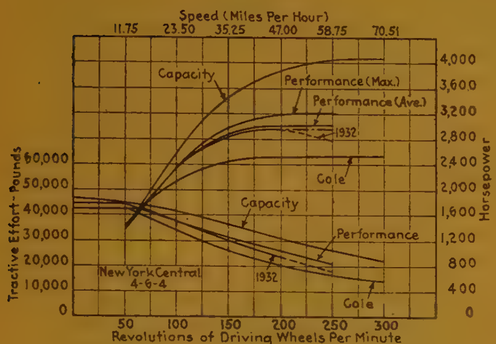

Fig. 10.

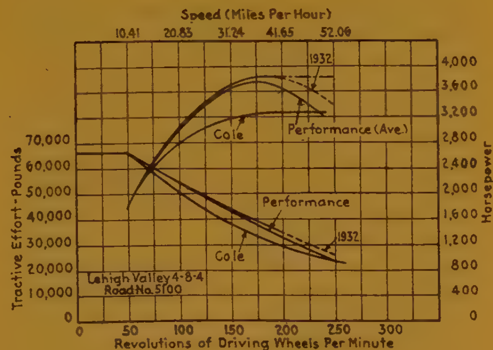

Fig. 13.

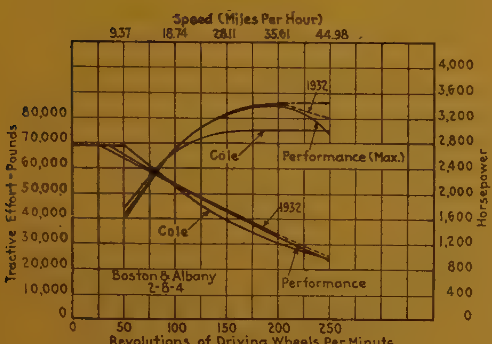

Fig. 11.

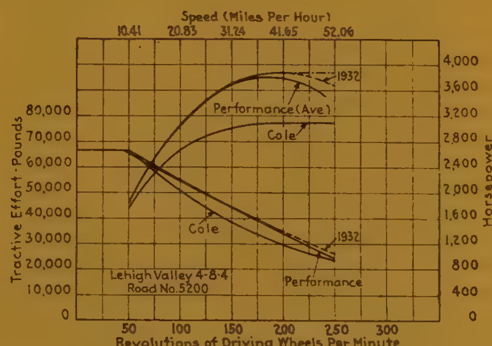

Fig. 14.

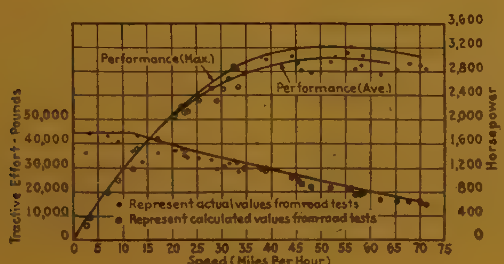

Fig. 15.

figures in circles represent the ratios of heating surface to grate area for each of the five locomotives. In other cases they have also been very high, indicating that Cole's evaporation corresponds very closely to a rate of firing of 120 lb. per sq. foot of grate area per hour, as was assumed by him.

The shape of the curves evidently depends upon the ratios between the principal dimensions of the boilers. It is not possible to find a formula for the curves of evaporation as functions of these variables. They do not run consistently enough, and there are other variables that cannot be put in a formula, as, for instance, the quality of coal. More consistent results were obtained from plotting curves of road tests, as shown in figure 2. Three locomotives are considered here (New York Central J-1a and L-2, and Boston & Albany A-1), representing three different classes of service : High-speed passenger, high-speed freight, and moderate-speed freight.

As stated above, there are no test data which would permit plotting these curves directly. An indirect method was therefore evolved for this purpose. From road tests with locomotives the maximum-performance indicated horsepowers and corresponding speeds and cutoffs were chosen. From the Altoona test, locomotives of similar engine size were selected, and the steam consumption per horsepower was found. It was corrected for difference in boiler pressures and superheats, and further, a certain percentage for auxiliaries was added from road-test data. Thus the probable specific consumption of road locomotives per indicated horsepower at various speeds was established. The product of horsepower and the established specific steam consumption furnished a means of finding the probable steam evaporation of the road locomotive for the maximum-performance power.

It will be seen that in this case the variation of the maximum evaporation of different locomotives is much less; it is between 2.7 and 6.7 % greater than the Cole figure. The variation in the location of the peak points in relation to speed is also less, and they all happen to be between 150 and 170 r.p.m. Nevertheless, even in this case it was not possible to tie up the shape of the curves with the principal dimensions of the corresponding locomotives by a mathematical formula.

Other locomotives were also studied, and on the basis of the accumulated information it was found possible to draw a probable curve of evaporation, as shown in figure 3. It has a peak point 7 % above that of the Cole figure, and at about 200 r.p.m.

All locomotives shown in figure 2 were equipped with feedwater heaters, and it is very striking to find that the average maximum evaporation is about 7 % above the Cole figure. This is easily explained by the increase in efficiency due to feedwater heating. In other words, we may assume that the Cole evaporation figures still hold good for boilers on modern locomotives without feedwater heaters, but when locomotives are equipped with feedwater heaters, the boilers generate 7 % more steam.

It will be seen that the curve was plotted rather conservatively : at low speeds it follows the lowest portions of curves from road tests, while at high speeds it is slightly above the average for road tests—this for the reason that other locomotive data with feedwater heaters pointed in this direction. The plotting of this curve can be done by using the following formula :

$$E = \beta E_c \quad \ldots\ldots\ldots\ldots \quad (7)$$

where E = boiler evaporation at a certain speed.
E_c = Cole evaporation figure
β = evaporation coefficient in relation to the Cole evaporation figure.

The evaporation coefficients are shown in figure 3 and repeated in table 1 in relation to the crank speed.

It may appear that the average curve was plotted rather arbitrarily, and that another curve could be chosen for the average evaporation with equal justification. It will be seen later that the ultimate recommendations for plotting horsepower and tractice-force curves are based on test results and do not depend directly on the evaporation coefficients. They are independent of the evaporation curve plotted in figure 3, which is given here simply as an illustration of the principle of the method.

TABLE I. — **Evaporation coefficients.**

	Evaporation coefficient	
Speed r. p. m.	Locomotives without feed-water heater.	Locomotives with feed-water heater.
50	0.60	0.65
100	0.85	0.91
150	0.98	1.05
200	1.00	1.07
250	0.93	1.00

Recommended method.

When the boiler evaporation is determined, the indicated boiler horsepower can be found from the formula (6):

$$P_i = \frac{E\,(1 - x)}{S_h} \quad \ldots\ldots\ldots \quad (6)$$

where E = evaporation, lb. per hour,
x = ratio of steam used for auxiliaries,
S_h = steam consumption per horsepower-hour, lb. (steam rate)

provided S_h is known.

It was thought more advisable to study $S_h / (1 - x)$, which represents S_a, the steam consumption per horsepower-hour, including auxiliaries, rather than S_h and x separately. Various locomotives were analyzed and it was found that the average steam consumption per horsepower-hour, including auxiliaries, can be well represented by one of the two curves shown in figures 4 and 5, depending upon whether we refer them to the r.p.m. or to the piston speeds.

It can be seen that the two curves differ very little; in other words, the piston speed has actually very little effect on steam consumption within the limits of strokes in modern locomotives, namely, 28 to 32 inches. It is therefore practically of little importance for the computation of boiler horsepower whether the steam rate is taken as a function of crank speed or piston speed, at least for strokes between 28 inches and 32 inches.

The method of figuring boiler horsepower can thus be reduced to the following :

On the basis of general boiler dimensions the Cole evaporation E_c is figured and the various evaporations E for different crank speeds n (r.p.m.) are calculated in accordance with formula (7) and table I. The figures thus obtained for E are divided over the corresponding steam rates S_a of figure 4 and the quotients obtained are horsepowers P_i for the various speeds.

If it is desired to use figure 5, which gives steam rates with respect to piston speeds, then it is necessary that the evaporation figures estimated in accordance with formula (7) and table I be referred to piston speeds, which can be calculated for various crank speeds n by the formula

$$S_p \; \frac{s = n}{6} \quad \ldots\ldots\ldots \quad (8)$$

where S_p = piston speed, feet per minute,
s = stroke, inches.

In either case, whether the horsepower has been figured in relation to crank speed or piston speed, in order to find the tractive force, it is necessary to refer horsepower to locomotive speed, which can be done by using the formula

$$V = \frac{D \times n}{336.134} \quad \ldots\ldots\ldots \quad (9)$$

The tractive effort is easily calculated by using formula (5).

Thus the boiler tractive force is determined and can be plotted on a chart. The rated tractive force is then figured, plotted as a straight constant-force line on the same chart, and extended until it intersects the boiler tractive force. The point of intersection determines the speed at maximum tractive force.

Regarding steam-rate curves (figs. 4 and 5), the following is to be said :

The test data were not sufficient for plotting through the whole range of speed. Only a portion of the curves, representing the most frequent average speeds of road tests, could be plotted. In order to cover the whole range of speed, an inverted method has been used; assuming that the coefficients of evaporation β as given above were correct, the total evaporations of several existing locomotives were calculated as functions of speed and divided over the corresponding horsepowers. The results for various curves are shown on figures 6 and 7 in reference to r.p.m. and piston speed. Figures 4 and 5 represent averages of the curves shown on figures 6 and 7.

It will be seen that the steam rates S_a give very consistent results at low speeds up to about 150-180 r.p.m., while above 200 r.p.m. the steam rates vary rather widely. This cannot be due to the coefficient of evaporation β, because this is practically constant between 150 and 250 r.p.m. It is due to the usual discrepancy between locomotive-test results at high speeds, because the influence of such factors as quality of coal, size of nozzle, drafting arrangement, steam passages in the cylinders, and valve-motion design become more pronounced at higher speeds.

The average curves check fairly well with steam-consumption figures obtained from tests for the ranges of speed for which data are available. By correcting them back through the coefficient of evaporation and test results for horsepower, the influence of β is eliminated if the horsepower of an existing locomotive is calculated as suggested above, on the basis of β and of steam rate S_a corrected on the same basis. This will be made still clearer later, when the simplified method is described.

An example may better illustrate the method. Suppose that we wish to find the tractive-effort curve for a 4-6-4 locomotive with 25-inch by 28-inch cylinders, 225 lb. per sq. inch working pressure, 79-inch driving wheels, Type E superheater, feedwater heater and a boiler of given dimensions (New York Central J-1a).

We determine first the Cole evaporation E_c in accordance with the *A. L. Co. Handbook* or calculate the evaporation by some other means—for instance, assuming an average evaporation of 12.2 lb. per sq. foot of total evaporation heating surface. If we follow the *A. L. Co.* figures, we find that E = 54 662 lb. of steam per hour.

Suppose that we prefer to figure on the basis of steam rate in relation to piston speed rather than r.p.m. In this case draw up table II on the basis of the curve, figure 5. Then plot the boiler tractive-force curve *DE* (fig. 8). Calculate the cylinder (starting) tractive force and plot a horizontal line *AD* corresponding to the cylinder tractive force up to its intersection with the boiler tractive force. The line *ADE* represents the indicated tractive force of the locomotive calculated on the basis of piston speed.

If instead of figure 5, figure 4 were followed, i.e., with the steam rate referred to crank speed in r.p.m. instead of piston speed, the procedure would be simpler, because no conversion to piston speed would be necessary, as can be seen from table III.

The results are plotted on figure 8 as

curve *BC*, and *ABC* represents the indicated tractive force of the locomotive calculated on the basis of crank speed. It can be seen that the difference between curves *ABC* and *ADE* is very slight.

TABLE II.

Cole evaporation (E_c)	54 662 lb. per hour				
Revolutions per minute (n)	50	100	150	200	250
Coefficient of evaporation (β)	0.65	0.91	1.05	1.07	1.00
Total evaporation (E)	35 530	49 742	57 395	58 488	54 662
Piston speed, ft. per min.	233	466	700	933	1166
Steam rate, lb. per I.H.P.-hour	25.4	21.3	20.3	19.9	19.7
Boiler horsepower, indicated (Pi)	1398	2335	2827	2939	2774
Speed, m. p. h. (V)	11.75	23.50	35.25	47.00	58.75
Boiler tractive force, indicated, lb.	44 616	37 260	30 074	23 449	17 706

TABLE III.

Cole evaporation (E_c)	54 662 lb. per hour				
Revolutions per minute (n)	50	100	150	200	250
Coefficient of evaporation (β)	0.56	0.91	1.05	1.07	1.00
Total evaporation (E)	35 530	49 742	57 395	58 488	54 662
Steam rate, lb. per H.P.-hour (Sh)	25.0	21.1	20.2	19.8	19.6
Indicated horsepower (Pi)	1421	2357	2841	2954	2788
Speed, m. p. h. (V)	11.75	23.50	35.25	47.00	58.75
Indicated tractive force (Ti)	45 351	37 611	30 223	23 568	17 795

Simplified method.

It can be seen that if the above-described method is carried out on the basis of crank speed in r.p.m. (table III), the calculation can be simplified by combining various constants into a single one, and obtaining the indicated-horsepower and tractive-effort figures directly. We already have the following relations :

$$P_i = \frac{E(1-x)}{S_h} = \frac{E}{S_a} \quad \ldots \quad (6a)$$

$$E = \beta E_c \quad \ldots \quad (7)$$

$$T_i = \frac{375 P_i}{V} \quad \ldots \quad (5i)$$

$$V = \frac{Dn}{336.134} \quad \ldots \quad (9)$$

Consequently

$$P_i = \frac{\beta}{S_a} E_c \quad \ldots \quad (10)$$

and

$$T_i = \frac{375\beta \times 336.134}{S_a \times n} \frac{E_c}{D} = \frac{126.050 \times \beta}{S_a \times n} \frac{E_c}{D} \quad (11)$$

or denoting the factors preceding E_c in (10) by M_p, and preceding E_c/D in (11) by M_t, we may write

$$P_i = M_p \times E_c \quad \ldots \quad (12)$$

$$T_i = M_t \times \frac{E_c}{D} \quad \ldots \quad (13)$$

The values M_p and M_t may be called the « horsepower modulus » and « tractive-force modulus, » respectively, and they may be calculated from the above definitions, namely,

$$M_p = \frac{\beta}{S_a} \quad \ldots \quad (14)$$

$$M_t = \frac{126.050 \times \beta}{S_a \times n} \quad \ldots \quad (15)$$

Coefficient β and the steam rate S_a are functions of the crank speed n; therefore M_p and M_t are also functions of n and may be calculated on the basis of the previously stated values for these variables (table I and fig. 4). They are given table IV.

Thus the calculation of horsepower and tractive effort is reduced to the operations indicated in table V, the results being practically identical with those of table III.

If the tractive force is the only curve which is desired and the horsepower values are not immediately wanted, the calculation can be still simplified by figuring only T_i and using the modulus M_t.

TABLE IV.

Revolutions per minute (n)	50	100	150	200	250
Modulus $Mp \times 1000^{19}$	26.00	43.13	51.98	54.04	51.02
Modulus Mt (See footnote 3)	65.55	54.36	43.68	34.06	25.72

As an illustration of the simplified method, the previously mentioned locomotives, for which reliable road-test results are available (New York Central L-2, J-1a, Boston & Albany A-1, Timken locomotive, and Lehigh Valley locomotives Nos. 5100 and 5200), have been checked and the results plotted on the charts of figures 9 to 14 in relation to crank speed and locomotive speed in miles per hour. The performance curves previously referred to are drawn on all these charts, and the corresponding simplified-method curves are marked « 1932. » The Cole curves are also given for comparison, and on two locomotives the capacity-test curves are also shown. It will be seen that the agreement is sufficient for practical purposes, and that the simplified method gives the maximum performance figures very closely. In the case of the New York Central J-1a and Timken roller-bearing locomotives, the discrepancy shown may have been due to the good quality of coal used in the tests.

TABLE V.

Cole evaporation (E_c)	54 662 lb. per hour.				
Revolutions per minute (n)	50	100	150	200	250
$M_p \times 1000$	26.00	43.13	51.98	54.04	51.02
$P_i = M_p \times E_c$	1421	2357	2841	2954	2788
M_t	65.55	54.36	43.68	34.06	25.72
$T_i = M_t \times E_c / D$	45 354	37 611	30 222	23 566	17 795

The 1932 curves are drawn in solid lines up to 200 r.p.m. The agreement is very good, because the test results, as has already been pointed out, are very consistent up to that speed. For average conditions the curve will most probably have, at least for performance tests, a drooping characteristic, as shown for curves 1932, but it is not impossible that it might approach a horizontal line, shown by dots and dashes. More experimental data at high speeds will be necessary in order to impart more definiteness to that portion of the 1932 curve.

It has been pointed out earlier that the ultimate figures for horsepower and tractive effort do not depend upon the evaporation coefficient. This can be

(3) These moduli apply to locomotives with feedwater heaters; for locomotives without feedwater heaters they should be reduced by 7/107 = 6.54 %.

more clearly seen from formulas (14) and (15), in which the ratio of S_a to β is what matters. The steam rates S_a as plotted in figure 6 are, as has been explained in the proper place, figured on the basis of the evaporation coefficients of table I, and therefore their ratio does not depend upon the numerical value of either of the two.

There is no doubt in the author's mind that for locomotive horsepower, the boiler evaporation is the controlling factor, at least for the porportions found in existing locomotives. However, if we imagine a boiler of very large proportions, one able to supply more steam than we usually get in the most modern locomotives, longer cut-offs would become possible, and at high speeds the mean effective pressures might drop below those that are obtained at present for corresponding cut-offs, on account of the steam resistances in passages (wire drawing). It is doubtful whether this would happen on locomotives of conventional proportions, but as a safeguard it may be advisable to set a limit for the possible increase in maximum horsepower compared with Cole's cylinder horsepower, which for superheated steam in expressed by 0.0229 times the area of one cylinder in square inches times the boiler pressure in pounds per square inch [4].

In the author's opinion, that limit could be made not less than 24 % for engines with feedwater heaters, but as a conservative figure, 20 % should be recommended until further information can be gathered.

Of locomotives for which the charts of figures 9 to 14 show horsepower and tractive-effort curves, only one, the Lehigh Valley No. 5200, had a boiler sufficiently large to show an increase over the Cole formula of more than 20 %.

[4] *American Locomotive Company Handbook*, 1917, p. 54.

Principal dimensions of locomotives.

Railroad	New York Central.	New York Central.	Lehigh Valley.	Timken.	Boston & Albany.	Lehigh Valley.	Pennsylvania.	Pennsylvania.	Pennsylvania.	Pennsylvania.	Pennsylvania.
Class, or serial, number	J-1a	L-2	5100	1111	A-1	5200	K-4s	E-6s	K-2sa	L-1s	I-1s
Wheel arrangement	4-6-4	4-8-2	4-8-4	4-8-4	2-8-4	4-8-4	4-6-2	4-4-2	4-6-2	2-8-2	2-10-0
Cylinder diameter, inches	25	27	27	27	28	26	27	23 1/2	24	27	30 1/2
Cylinder stroke, inches	28	30	30	30	30	32	28	26	26	30	32
Wheel diameter, inches	79	69	70	73	63	70	80	80	80	62	62
Boiler working pressure, lb. per sq. inch	225	225	250	250	250	255	205	205	205	205	250
Weights in working order, lb :											
On drivers.	182 000	244 000	270 000	264 000	248 200	268 000	202 880	133 100	179 900	235 800	352 500
Total locomotive	343 200	364 000	408 000	417 500	385 000	422 000	390 140	240 000	293 200	315 600	386 100
Heating surfaces, sq. feet :											
Firebox	281 [1]	354 [1]	490 [1]	474 [1]	337 [1]	508 [1]	306.8 [1]	232.7 [1]	208 [1]	301.5 [1]	287 [1]
Tubes and flues	4203 [1]	4095 [1]	4932 [1]	4637 [1]	4774 [1]	4933 [1]	3728.4 [1]	2643.9 [1]	3451 [1]	3728 [1]	4487 [1]
Total evaporative surfaces	4484 [1]	4449 [1]	5422 [1]	5111 [1]	5110 [1]	5441 [1]	4035.2 [1]	2876.6 [1]	3659 [1]	4029.5 [1]	4774 [1]
Superheating surface, sq. feet.	1951 [1]	1938 [1]	2256 [1]	2157 [1]	2111 [1]	2243 [1]	1171 [2]	810.6 [2]	989.32 [2]	1171.63 [2]	2410 [2]
Superheater type	E	E	E	E	E	E	A	A	A	A	E
Grate area, sq. feet	81.5	75.2	88.3	88.3	100	88.3	70	55.1	55.4	70	70

[1] Wetted side. [2] Fire side.

Curve 1932 of figure 14 was therefore drawn in such a way as to have a peak point only 20 per cent above the Cole figure.

As was pointed out at the beginning, the object of this paper is to suggest a simple method for figuring horsepower and tractive force for modern locomotives. To this class belong locomotives with type « E » superheaters, feedwater heaters, and valve motions with about 8 1/2 inches of valve travel. This presupposes that the superheater heating surface assures sufficient superheat, which in locomotives with the type « E » superheater is about 250° F. If locomotives have higher superheats, the power may increase, but it was not thought advisable to give constants for various superheats for the reason that before a locomotive is tested, its superheat is not known, and therefore these constants would not be helpful in calculating the horsepower of a locomotive beforehand. It would be logical to base such constants on the relation of the superheating surface to the evaporative heating surface, or to the amount of generated steam, but a comparison of these ratios with test results of various locomotives had not been conclusive.

For locomotives of older design, with type « A » superheaters, new constants could be worked out similar to those given in the paper, but it is suggested that for these latter locomotives the Cole formula should be used. Until the new method has proved to be practical, the Cole formula should be used for locomotives with type A superheaters. This should not imply, however, that type A superheaters can never develop the horsepowers recommended by the new method. Locomotives are known that have given very high performance figures with type « A » superheaters on good coal.

***

DISCUSSION [5].

R. Eksergian (Engineering Department, E. I. du Pont de Nemours. Inc., Wilmington, Del.). — The author has pointed out the many difficulties in predicting accurately locomotive tractive force and horsepower characteristics, such as on the cylinder side of estimating and allowing for cylinder condensation and wiredrawing against speed, and on the boiler side the difficulties of estimating evaporation and likewise its variation with speed. His recommended method for the calculation of horsepower and tractive force against speed is unquestionably rational and probably the best approximation in a preliminary analysis. For estimating the evaporation the Cole method has been retained, with a correction coefficient as a function of the speed.

In the application of the author's method we are immediately confronted with allowances that must be made for variations in boiler proportions and for different types of locomotives. These variations become sufficient to question the need of the Cole evaporation figure. This does not mean that the data supplied by Cole on evaporation yields for firebox and particularly for tubes of different lengths are not of extreme value in aiding in the calculations of the correct evaporation of a boiler, but such data became modified, with large grates, combustion chambers, etc., so that the data are perhaps antiquated except for a guide in a first approximation to modern power. The author himself has seen the necessity of this in his correction coefficient

---

(5) Mr. Lipetz' paper, the several discussions and the author's closure appeared in full in the Transactions of the American Society of Mechanical Engineers, Railroad Division, Vol. 55, No. 9, Paper RR-55-2, pages 5 to 42 inclusive. The abstract presented here does not include the discussion contributed by H. S. Vincent, formerly chief consulting engineer, Franklin Railway Supply Co., nor that part of the author's closure relating thereto. The methods proposed by Mr. Vincent in his discussion is the subject of an article which also appears in this issue of the *Bulletin*.

The author has pointed out that the analytical method of estimating performance becomes extremely involved due to allowance for cylinder cooling, wiredrawing, etc., and on the boiler draft considerations, and the effect of grate, firebox, and other proportions on the evaporative yield.

George W. Armstrong (Consulting Mechanical Engineer, Ridgewood, N. J.) — The Pennsylvania test-plant experience for the locomotives used by the author represents boiler output at the time deemed capacity, with front-end design generally superior to most in use at that time. Subsequent improvements in front-end design have given much greater maximum capacity output. This naturally affects the engine or tractive-force output of the locomotive, and developments for improvement in draft conditions and brack-pressure reduction have in more than one instance resulted in 200 to 250 H.P. increased output, due to greater boiler capacity. The great value of front-end improvement lies in the ability to support more efficiently high combustion rates, with consequent increase in available steam for engine output.

Evaporations as high as 10 lb. of water per sq. foot of equivalent heating surface have been attained in road operation. On such a basis the New York Central J-1, cited by the author, should be capable of delivering around 78 000 lb. of steam per hour, instead of the 54 600 lb. which would materially influence tractive force and horsepower output. Experience with improved draft appliances on locomotives with boiler characteristics similar to this one indicates that this expectation is not unreasonable. The author has recognized this to some degree by placing « the peak point only 20 % above the Cole figure ».

W. F. Kiesel, Jr. (Mechanical Engineer, Pennsylvania). — The stated object of the paper is to establish a method of figures for calculation of indicated tractive-force as a function of speed. Throughout the paper references are made to the difficulties of establishing accurate formulas applicable to various types and variations of locomotives and detail equipment, establishing the need of adopting empirical data.

The author points out that Cole's ratios are no longer applicable to modern locomotives and submits his method of corrections necessary to make the Cole formulas more closely conform to modern power results.

The question of first importance is « How much steam can any given locomotive make available for use in the cylinders ? » The author is rather vague on this point. He calls attention to boiler tractive force as distinguished from rated tractive force. The latter is dependent on boiler pressure and proportions of cylinders and drivers and its calculation has been standardized by a simple formula. The designer is mainly interested in maximum boiler tractive force, which is dependent on maximum evaporation possibilities of the boiler. Conventional fire-tube boilers, with staybolted firebox, vary little in important proportions.

It is not far wrong to assume that the evaporation per square foot of combustion-space heating surface, compared with that of the flues and superheater (with flue sheets spaced about 20 feet apart) is 6 to 1. Assuming that equivalent heating surface is the sum of superheater, flue, and six times the combustion-space heating surfaces (steam and water side), it has been developed from tests that the evaporation limit to date is closely 11 1/2 lb. per hour per sq. foot of equivalent heating surface, which permits the empirical assumption that the steam available for use in cylinders may be as high as 10 lb. per hour for each sq. foot of equivalent heating surface, for the whole range of boiler tractive force. When test results of a locomotive fall materially short of this it is advisable to make an investigation to determine reasons for the shortage.

The second phase of this problem is to determine the results in cylinder tractive force of the use of various amounts of steam in the cylinders up to the available limit. The author determines drawbar horsepower, but it would seem preferable to find the cylinder tractive force and subtract engine and tender resistance, the result being drawbar pull at rear of tender. Piston speed, or crank speed, suggested in the paper, can be used, but the writer prefers to use speed V in miles

per hour, because that value is more generally used. He also prefers to determine cylinder tractive force T which can readily be transformed to indicate horsepower, if that figure is desired.

Assume:

Initial pressure P = 10 lb. less than boiler pressure,
Engine constant $C = d^2S/D$,
Steam constant $M = 3W/110w$,
d = cylinder diameter, inches,
S = piston stroke, inches,
D = driver diameter, inches,
W = amount of steam used, per hour,
w = average weight of steam per cu. foot at 100° superheat and pressure P.

Since modern locomotives, with few exceptions, use boiler pressures between 200 and 300 lb. per sq. inch the steam constant M may be written $M = xW$, the values of x being

Boiler pressure:	200	225	250	275	300
Coefficient x:	0.0716	0,0643	0.0571	0.0532	0.0493

Having determined P, C and M, the formula for tractive force becomes

$$T = \frac{2PM}{(M/C + V)}$$

The writer submits that this formula in use for the last 20 years, though to a certain extent empirical, is based on a rational theory, gives fairly close results and permits deriving more information to determine preferential locomotive designs with less effort than by the use of modified Cole ratios.

Capacity versus maximum performance.

G. T. Wilson (General Equipment Inspector, Motive Power, New York Central Lines, New York). — The writer thoroughly appreciates the fact that the results as obtained are subject to a factor of correction where the performance of the locomotive to which it is applied varies to an appreciable extent from the average results upon which the method of calculation is based.

The application of this boiler-performance calculating method to two of the most modern types of New York Central locomotives—the J-1, 4-6-4, and L-2, 4-8-2—indicates that the results as shown for tractive force and indicated horsepower in respect to speed are not representative of capacity rating for these locomotives as shown from dynamometer road tests.

It is the practice of the New York Central to rate a locomotive on capacity test results represented by the drawbar pull-speed curve for the respective locomotive or class. The capacity test results represent the maximum sustained drawbar pull and horsepower for all speeds for the respective class of locomotive as governed by the existing boiler ratios, drafting arrangement, valve setting, and cylinder characteristics.

For the past 20 years the capacity results have been used for tonnage rating with success. In the last few years we have made it possible by means of a device applied to the locomotive to provide the engineman with a visible indication of cut-off correlated to speed so that with full-open throttle the engineman may select a cut-off to produce maximum drawbar pull for that incidental speed. The cut-off indication of this device is based upon the capacity test results for that class of locomotive to which it is applied. The incidental cut-off in terms of speed corresponds to the cut-off used when maximum sustained capacity was developed during dynamoter road tests.

Experience has proved that we may duplicate in everyday performance the actual capacity test results by selection of the same incidental cut-off as used to develop the drawbar-pull-speed curve.

Based upon our observation from dynamometer tests and average everyday operation we do not believe that the results for tractive force and horsepower for a modern design of locomotive may be consistently based upon average performance tests as representative for the tractive force and power of the loco-

motive. We contend that the capacity performance of a locomotive is representative of the true characteristics of the boiler and engine because such results eliminate the human variable, common in performance results, and show the true performance as governed by the design of the boiler, feedwater heater, superheater, drafting arrangement, and steam distribution to cylinders.

A. Giesl-Gieslingen (New York). — The writer would like to comment on Mr. Wilson's remarks dealing with locomotive « capacity », versus « maximum performance ». Certainly a railroad is interested in the utmost limit of capacity of a locomotive, since it is of importance on frequent occasions. However, this limit is subject to many influences, the most imponderable of which is probably the boiler capacity as it is governed by the accidental quality of the coal and the ability of the fireman. These latter influences are, of course, always felt, whether the engine is working light or hard, but for every locomotive there is a reasonable maximum of performance which can safely be developed, whoever may handle it and whatever conditions may be, and this is what the author primarily determines. The boiler is then called upon to work only at a conservative rate.

Determination of horsepower.

Ching Pong Pei (Champaign, Ill.). — It is correctly stated in the paper that no one formula can express the tractive force of all locomotives. It is equally difficult to express all the factors affecting the development of the tractive force of the locomotive under the varying operating conditions in any single, simple formula or equation. Such being the case, in the formulation of any one simple equation for the tractive force of all locomotives one is forced to make a choice in the selection of the more important items and the elimination of the lesser ones.

The author, in arriving at a basis upon which his suggested method of calculation is constructed, used the most direct method of first determining the horsepower output of the locomotive and then obtaining the tractive force from the well-known equation $P = (T \times V) - 375$ or $T = (P \times 375)\ /V$.

The horsepower output of the locomotive is to be determined by the direct method of dividing the total boiler evaporation by the steam consumption of the locomotive cylinders and the auxiliary devices. The relation between the horsepower and the tractive force of the locomotive as defined by the latter equation is mathematically exact. Hence the whole problem is reduced to the determination of horsepower which entails only two factors — the total boiler evaporation and the total steam consumption.

The justification of correlating the boiler evaporation with the rotative speed of the locomotive, as expressed by the formula $E = \beta Ec$ is, as the writer sees it, premised on these assumptions:

1. The Cole evaporation figures E as derived from the performance of the locomotives of the pre-war period still hold good on modern locomotives.

2. The draft efficiency is uniform on all locomotives, being only a function of the amount of steam passing through the exhaust nozzle, which implies that the same amount of steam is exhausted from the cylinders at a certain rotative speed of all locomotives.

3. The firing rate in the fire box and the boiler efficiency are each only a function of the draft produced in the front end.

There is no way to prove or disprove the validity of the first assumption, since no mention has ever been made about the firing rate in connection with the Cole evaporation figures. Likewise, the author avoided specifically stating the firing rate or the boiler efficiency at which the evaporation figures were derived. There is no doubt that at some firing rate an evaporation figure as shown by the Cole figure can be obtained. It is equally true at some other firing rates that the evaporation rate can be either higher or lover than the Cole figures. In the derivation of boiler-evaporation figures the author stated that the Cole evaporation figures still hold good on modern locomotives. Whatever significance may be attached to the Cole evaporation figures, it

is clear from the foregoing statement according to the author, that there has been no material improvement in the locomotive boiler performance, with the exception of the increase in boiler capacity made possible by the addition of feedwater heater, during the last 15 or 20 years. Does this not offer a challenge to the railroad mechanical department in general and the locomotive designer in particular ?

It is rather difficult to subscribe to the second assumption in that the same amount of steam is always exhausted from the cylinders at a certain rotative speed of the locomotive. This would be true only when the locomotive is operated with a definite and exacting relationship between the cut-off and the rotating speed. In a general way, the locomotive is operated with longer cut-offs at low speed and with shorter cut-offs at high speeds, and this is as far as the relation between the cut-off and the rotative speed goes. Hence, this assumption cannot hold true for any one locomotive, let alone for all locomotives.

The steam consumption of the locomotive cylinders is naturally mainly a function of the percentage of cut-off at which the engine is operated. It is also a function of the pressure and the temperature of the steam, both at the steam chest and at the point of cut-off. Test results of locomotives on testing plants seem to indicate that there is always a narrow range of cut-offs to be operated with a certain rotative speed, the combination of which results in the minimum steam consumption, other conditions remaining unchanged. On the other hand, if we were interested only in the maximum capacity of the locomotive it may be operated with a much longer cut-off than that at which the minimum steam consumption is obtained, limited only by the capacity of the boiler to supply steam. At any stated boiler capacity, which is mainly a function of the amount of fuel burned, the locomotive can be operated with a number of different cut-offs at one rotative speed, in which case the locomotive would develop different values of tractive force, according to the cut-off, at the same rotative speed. It is, therefore, not sufficient merely to correlate the tractive force and speed, as represented by the ordinary tractive-force-speed curves, without specifying the manner in which the particular tractive force figure is obtained.

It is not possible to express the tractive force of a locomotive by any single, simple formula, however desirable it may be, and it is also almost meaningless merely to state that the locomotive would develop so many pounds of tractive force at a certain speed without specific qualifications. In view of these difficulties, would it not be more logical to present the locomotive tractive-force data in the form of a series of tractive-force-speed curves, each one representing a certain firing rate and each point on a curve representing the tractive force obtained with the most suitable cut-off at the corrspeonding rotative speed ?

The question of superheat.

John E. Muhlfeld (Consulting Engineer, New York). — Concerning the author's proposed modernization of the Cole ratios for the purpose of meeting present-day requirements, the writer questions the differentiation based on the so-called types E and A superheaters. In his conclusion the author states: « The object of this paper is to suggest a simple method for figuring horsepower and tractive force for modern locomotives. To this class belong locomotives with type E superheaters, feedwater heaters and valve motions with about 8 1/2 inches of valve travel. This presupposes that the superheater heating surface assures sufficient superheat, which in locomotives with the type E superheater is about 250° F. For locomotives of older design, with type A superheaters, new constants could be worked out similar to those given in the paper, but it is suggested that for these latter locomotives the Cole formula should be used. »

Any number of modern steam locomotives equipped with type E superheaters, with from 200 to 250 lb. boiler pressure, are not averaging superheated steam of over 600°. F., or much over 200° of superheat, when working steam. This performance can be duplicated by many of the older locomotives originally

constructed without superheaters, but which since have been equipped with type A superheaters, and, with 200-lb. boiler pressure, can raise total steam temperatures of from 700° to 750°, and average from 600° to 650° F., thereby producing as high as 350° superheat as a maximum and from 200° to 250° F. as an average.

The particular feature of the type E superheater has been its increased superheating surface which reflects favorably on the boiler capacity and, in combination with the higher velocity draft, has been set up as the principal advantage over the type A. However, as compared with the type A the type E has many operating disadvantages owing to the overheating, swelling and burning out of the torpedo type of forged return bend; the more restricted gas area through the superheater flues; the stopping up with cinders, ash, soot and other foreign matter of superheater flues resulting in stuck units; the increased gas velocities through the superheater flues tending to cut out crownsheet staybolt heads and the beads on the firebox end of the flues, and to a more rapid cutting action of the cinders against the return bends and the element necks, necessitating application of innumerable shields to these parts to prevent such action. All of these conditions mean increased maintenance troubles and expense and of operating inefficiencies and failures which do not obtain with the type A superheaters.

Summing the locomotive superheater situation, it is the writer's opinion that the author's method of calculating tractive force should be used on locomotives having 100 % or larger boilers, modern long-travel valve gear, and a sufficient degree of superheat, whether equipped with a type E or a type A superheater.

With respect to high superheat temperatures, it is questionable as to the practical advantage or economy in superheating steam to higher temperatures than what is required for minimum cylinder condensation, in view of the useful heat that will be exhausted to the atmosphere. To produce 70 000 lb. tractive force in a single-expansion-cylinder locomotive carrying 200 lb. pressure, with 63-inch driving wheels, would require the admission of live steam into two 28 1/2-inch diameter by 32-inch stroke high-pressure cylinders. At 250 lb. pressure the diameter of the cylinders would be about 23 1/4-inches. If 500 lb. pressure is used in combination with single-expansion cylinders, it would only be necessary to put live steam into two 18-inch diameter by 32-inch cylinders, with 63-inch diameter driving wheels, in order to develop 70 000 lb. tractive force and, in which case, with the conventional amount of superheat, from 75° to 100° of superheat would be wasted in the exhaust steam.

By increasing the conventional 200 to 250-lb. boiler pressure, assuming 200° F. superheat as a constant, regardless of the pressure, it has been shown that only 28.5 additional B. T. U. are required to produce steam at 500 lb. as compared with 200 lb.

The author has contributed valuable information concerning calculations for designing modern steam locomotives which will be of great benefit to railroad engineers and locomotive builders. At the same time practical operating and maintenance factors should not be overlooked. These can only be determined by dynamometer-car tests and the data can be scientifically utilized to develop empirical formulas.

Feedwater heaters and front ends.

Thos. C. McBride (Consulting Engineer, Railroad Department, Worthington Pump & Machinery Company). — The system of locomotive ratios proposed by the author considers the feedwater heater and its effect in increasing the maximum evaporation obtainable from the boiler. The statement of this increase introduces a new and important ratio. A large amount of information that has been collected over a number of years on locomotives without heaters is made available in the consideration of present-day locomotives with heaters by the application of this new ratio. Careful consideration and accurate statement of this new ratio are, therefore, important.

The author reaches the conclusion that the Cole evaporation figures still hold good for boilers on modern locomotives without feedwater heaters, but when locomotives are equipped with feedwater heaters, the boilers gener-

ate 7 % more steam. In contrast with this conclusion, feedwater temperatures obtained from feedwater heaters on locomotives operating at the maximum rates of the Cole ratios indicate an average of approximately 15 % additional heat to the boiler through the heater, and this additional heat can get out of the boiler only through a like increase in evaporation. The difference between these two figures is too great to neglect, especially in developing a system of locomotive ratios.

The author has taken great pains to collect and carefully apply a mass of data obtained partly from road trials of locomotives, but road trials necessarily involve large « probable errors ». A few temperature readings taken simultaneously while the locomotive is known to be working at the desired maximum make it possible to calculate the extra heat going to the boiler through the heater with a probable error of but a few per cent. It is necessary only to read the temperatures of the water entering and leaving the heater and the steam pressure and superheat at the moment the indicator card or other check is made on the capacity of the locomotive. The process can be repeated every few minutes and more exact information gathered in an hour than could be obtained from a month of road trials.

The writer has shown that (assuming 200 lb. boiler pressure, 150° F. superheat, and 215° F. feedwater temperature obtained, with alternate assumptions of water at 40° F. from the tender to represent winter and 70° F. to represent summer conditions) the additional heat supplied to the boiler through the heater is 13.6 and 11.1 % respectively. That is, presuming exactly the same amount of fuel burned and exactly the same amount of heat transmitted to the water and steam in the boiler, with the consequent same efficiency of the boiler, the feedwater heater increases the heat to the boiler, and, consequently, the evaporation from the boiler by 13.6 and 11.1 % respectively, under the conditions assumed.

These assumed conditions were considered representative of usual road operation in 1920. When locomotives are operated at the maximum contemplated in the ratios, much higher feedwater temperatures are obtained, generally 240° F. and frequently 250° F.

If the conditions assumed in the paper of 1920 are again used for the sake of comparison, but with a feedwater temperature of 240° F., each pound of water will carry into the boiler from the heater 240 — 40, or 200 B. T. U., and will require 1284.6+32—240, or 1076.6 B. T. U. from the fire for generation into steam. The heater increases the heat from the fire by 200/1076.6, or 18.6 %. But of the total of 118.6, 2 %, or 2.4, is required to operate the feed pump, leaving 118.6 — 2.4 or 16.2 % net increase in heat because of the heater. Similarly, with the alternate assumption of water at 70° F. from the tender, each pound of water carries into the boiler 240—70, or 170 B. T. U., and requires from the fire the same 1 076.6 B. T. U. The addition to the heat from the fire by the heater is 170/1 076.6, or 15.8 %. Of the total of 115.8, the feed pump requires 2 %, or 2.3, for its operation, leaving 113.5 or a net 13.5 % of additional heat to the boiler through the heater.

The average of 16.2 and 13.5 is 14.8, or practically 15 %. It is thought that this figure, or some figure close to it, depending on the conditions assumed to represent average present-day locomotive operating conditions, should be adopted for the ratios, instead of the 7 % obtained from road trials.

Summing up, the writer believes we are bound to recognize the differential of 15 % or thereabout increased evaporation in favor of the boiler with heater, in view of the direct and accurate evidence of the temperatures obtained. It will then be necessary to modify figure 3 and table 1, spreading the « evaporation coefficient » to a differential of 15 % at high and medium r. p. m. But over all this looms the condition of the fire, new or old, demanding at least recognition.

H. B. Oatley (Vice-President in Charge of Engineering, Superheater Company). — The Cole ratios, which have been considered as a standard for more than a generation, were, and still are, of great value and have had widespread use both by railroads and locomotive builders. The marked change in value of factors entering into the design of locomotives during this time has made it necessary that extensions to and modifications of the Cole

ratios be made so that there could be comparisons between different types of locomotives and the effect of improvements be evaluated.

The paper offers methods and formulas which, while somewhat more complicated than have been used hitherto, are more easily applied and far less complex than methods used abroad. Any set of equations must, of necessity, be considered approximate within recognized limits. The interrelation between combustion, heat absorption, and conversion to external work is so intimate that the design of the various parts introduces a large number of variables that cannot be accurately determined.

W. A. Pownall (Wabash, Decatur, Ill.). — Figure 12 shows the Timken engine with a performance curve much higher than the Cole or the proposed method. The author states that this may be due to a good quality of coal, but it is believed that this exceptional performance at high speeds is due to the type of front end used on the Timken locomotive. The exhaust nozzle is a six-ported star-shaped design, of larger area than the customary round nozzle tip, but which at the same time produces a steam jet with greater entraining power due to its greater periphery. The usual draft plates and deflector plates in the front end are dispensed with, and in their place is an inside stack with flare, the lower edge of which is about 12 inches above the bottom of the smoke arch, and the netting consists of a straight cylinder extending from the lower edge of the stack to the bottom of the smoke arch. The stack itself is somewhat larger than the conventional stack. This arrangement produces very strong draft, and with less back pressure on the cylinders and because of the strong draft, grates with reduced air opening have to be used.

The Wabash has equipped a 4-8-2 type locomotive with similar front-end arrangement, and has obtained as high as 4 900 I. H. P. at a speed of 55 m. p. h. The maximum cylinder horsepower by the Cole method would be 3 215, and by the Lipetz method 3 237. This engine in actual service evaporated at times water at the rate of about 80 000 lb. per hour, whereas the maximum evaporation from the Cole method is figured with coal burned at the rate of 120 lb. per sq. foot of grate area per hour, whereas this rate was very much exceeded during the actual performance of the engine in question. Tests of Pennsylvania engines equipped with this front-end arrangement (this design of front end was developed on the Pennsylvania) have shown similar evaporative rates and increased horsepower. The high horsepower developed is undoubtedly contributed to by a lower water rate due to reduced back pressure, and the writer would point out this low water rate on the Timken engine as indicated in figures 6 and 7 of the paper.

Piston speed and speed factors.

L. K. Sillcox (Vice-President, New York Air Brake Co.). — The variation of speed factors with the piston speed for superheated locomotives are shown in table 15, as derived by Cole in March, 1910, at which time it was found that the average maximum horsepower was reached at 1 000 ft. piston speed per min. and remained constant at higher speeds. Recent tests with locomotives indicate a distinct rise in percentage of about 10 % beyond the figures shown from 1 000 feet piston speed and up, and herein lies the advantage which is observed in the locomotives constructed in 1933 compared with 1910, when sustained capacity at speed is to be judged.

TABLE 15.

Piston speed, feet per min.	Speed factor	Modified speed factor: 1910 (Cole) Modified factor = 0.85	Modified speed factor: 1933 (test) Modified factor = 0.93
0.250	1.000	0.85	0.930
300	0.955	0.812	0.888
500	0.770	0.655	0.716
700	0.605	0.514	0.562
1 000	0.445	0.378	0.414
1 200	0.371	0.315	0.345
1 400	0.318	0.270	0.296
1 600	0.278	0.236	0.259

One can readily subscribe to the accuracy of the author's method of approach to the problem of predetermining locomotive characteristics. So long as authentic data are available for guidance in estimating the probable relation between the expected and Cole evaporation, no marked discrepancies should be introduced in the results. Much of the familiar Cole formula is retained. In fact, the effect of the addition of a superheater might be similarly expressed by constructing a curve to indicate the relative performance of a locomotive so equipped with respect to one employing saturated steam. Conversely, the Cole constants may be extended to include factors which take into account the benefits derived by the preheating of feedwater, by the partial eliminating of throttling losses effected by long-travel valve gears, and by any improvement which serves to increase the availability of the energy contained in the fuel fired. In like manner the Cole speed factors may, by the application of proper modifications, be rendered applicable to any change in the relation between boiler and cylinder capacity, anticipating probable change in the shape of the horsepower curve.

Whether the author's method is adopted or the Cole analysis modernized, the results should bear marked similarity. Locomotives must be grouped, in either case, on a basis of relative proportions and the extent to which the latest refinements are incorporated in their design. Both are dependent upon the accumulation of adequate test data for the degree of accuracy experienced. While simplification of calculation is claimed for the author's analysis, familiarity, confidence and universal acceptance recommend retaining the Cole method in principle. The many tests already catalogued in their agreement with calculated performance using the Cole equations suggest the desirability of continuing the practice so well established in order that a common basis comparison of the old and the new may be erved. The introduction of any method represents the departure from an ac- standard will likely be strenuously l so long as the familiar process can ed to produce dependable results.

S. S. Riegel (Mechanical Engineer, Delaware, Lackawanne & Western). — We worked out the horsepower and tractive force curves for our most recently acquired fast-freight locomotives by both Mr. Lipetz's simplified and the Cole methods, and in our dynamometer tests, when the locomotives were first received, we developed horsepower and tractive force curves in harmony with the results secured by the Lipetz formula. We naturally are satisfied with the deductions as stated in the paper.

Author's closure.

At the beginning of the paper it was pointed out that there are two possible methods of evaluating the horsepower and tractive effort of a locomotive. In the first method, by using mathematical analysis, it would be quite possible to follow through the various processes taking place in the cylinders for a certain amount of steam of known quality (pressure and temperature). This analysis would be very intricate, as a great number of complex factors has to be taken into consideration. Furthermore, as such analysis would represent a consecutive chain of premises and sequences, the final result would depend upon the validity of each individual premise and there would be no assurance that the method would lead to reliable results.

In order to eliminate all these difficulties and offer a practical quick method for evaluating the horsepower of a locomotive, it was suggested in the paper to apply an empirical method, by which the power is figured on the basis of the total evaporation of the locomotive boiler and the approximate steam rate per horsepower-hour at various speeds. Thus only two variable factors are introduced in the calculation instead of a multitude of factors, and in addition both factors are such that they can be figured fairly accurately. The evaporation of a locomotive boiler is more or less known. It has been measured and evaluated hundreds of times, and the Cole figures seem to represent very closely the amount of steam which can be generated by a locomotive under normal conditions without excessively forcing the boiler. As to the steam

rate, it is also known that a well-proportioned and properly designed locomotive has fairly definite limits for steam consumption per horsepower-hour for various conditions of work. It would seem, therefore, that a method by which these two variables are tied together should give more accurate results than complex formulas.

Second, it must be borne in mind that the objective of the method is not to predict with accuracy the power-and-tractive-effort curve of any locomotive of any design. As Mr. G. W. Armstrong puts it rightly, the object is to provide a « comparometer » for evaluating locomotives of known designs. One, or another locomotive having an unusually large superheater, or firebox, or combustion chamber, or cylinders, or a special draft arrangement, or some other novelties which are being tried out from time to time, can not be evaluated by the suggested method. Only the average well-proportioned and properly designed locomotive can be served by this method with a reasonable degree of accuracy, and the locomotive with the enumerated or other improvements has to be evaluated by comparing its performance with the predictions of the formula.

Third, as it has been already pointed out in the paper, there are a great many kinds of tractive effort which are of interest and are being considered in diffrerent cases. It is therefore always possible to find fault with one or the other method of figuring tractive efforts and to point out that some tests show higher or lower figures than those obtained by the method. Tractive effort is not a definite thing, and it is not the *maximum* tractive effort which is being sought. It has been explained in the paper that the method permits to determine the *performance* curve which is usually obtained under normal conditions of work; and I should like to add here—and with a reasonable degree of efficiency. This will be explained later.

As regards the paper itself (not the method), it should be remembered that the theories and various statements made in the paper which may be open to criticism have no influence on the method in the final recommended form. Neither the theory of the variation of evaporation with speed, nor the evaluation of the variation of coefficient β, nor the actual steam rate as obtained from tests, has any bearing on the final results. These theories and statements had been brought out in order to make clearer the way by which the method was developed. After formulas (12) and (13) and table 7 had been established, they were checked for all modern locomotives for which test data were available, and the results were shown in figures 10 to 15 of the paper. The preliminary theories and statements were thus no more than stepping stones, which can be removed after the final formulas and empirical constants had been established and verified, and the criticism of these theories, by purely theoretical considerations, cannot undermine the method itself.

Mr. Kiesel does not think it necessary to figure the evaporation of a locomotive separately for various parts of the boiler as established by Mr. Cole. He recommends to figure the evaporation on the basis of *equivalent* heating surface, assuming the equivalent heating surface equal to the sum of superheater, flue, and six times the combustion-space heating surfaces, and the average evaporation equal to 11 1/2 lb. per hour per sq. foot of so-defined equivalent heating surface. The firebox heating surface seems to be a part of the combustion-space heating surface. This designation for the equivalent heating surface is based on an empirical law found by Mr. Kiesel—namely, that the evaporation of a combustion-space heating surface is six times as high as that of the flues and superheater, if the length of the flues is about 20 feet. Mr. Kiesel further assumes that 10 lb. per sq. foot of equivalent heating surface represents steam available for use in cylinders.

I do not think that such a rough method of calculating the evaporation would satisfy us, with all the knowledge that we have now, especially the inclusion of superheating surface into the total surface for figuring evaporation. This can be considered only as a necessary correction for the increase of power due to superheating, although one correction for the effect of superheating is already in-

cluded in the Kiesel formula by referring w to pressure P with 100° superheat. Moreover, such a method offers no advantages over the more accurate Cole formula, which has been proved to be satisfactory by many years of use, and to hold good even now for modern locomotives, as it has been shown in the paper and corroborated by many discussers. A simple calculation of evaporation for any of the locomotives cited in the paper, for which we have data, will show that Mr. Kiesel's empirical rule gives highly exaggerated figures. They are not evaporation figures any more, but simply values of W to be substituted in Mr. Kiesel's formula, which thus becomes a purely empirical formula. Mr. Kiesel concedes this, but is of the opinion that the formula is based on a rational theory. This may be true, but the rationality of it is more than offset by the omission of a great number of important factors which cannot be expressed by mathematical formulas, and therefore must be compensated by such figures as evaporation of a superheating surface and others.

The author cannot agree with the statement that he is vague on the point of how much steam any given locomotive can make available for use in the cylinders. There is a definite formula in the paper for the amount of steam available for the cylinders—namely, E $(1 - x)$, just preceding formula (6). This is made use of in formulas (6) and (6-A); evaporation E is explained and figured in the paper, x is included in figures 5 and 6. Mr. Kiesel's statement that the author « determines drawbar horsepower » while he prefers « the cylinder tractive force » is evidently a misunderstanding, as the paper is about the indicated horsepower and tractive effort, while hardly mentioning the drawbar horsepower at all.

The question of *maximum* versus *performance* tractive force was discussed by Messrs. G. T. Wilson and A. Giesl-Gieslingen. The latter's discussion may be considered as a reply to Mr. Wilson, which makes a further reply unnecessary. The author will only add that he stated in the paper his reasons for working out a method by which the performance, and not the maximum horsepower and tractive-effort curves, would be plotted. It would not be desirable to develop a method by which only capacity curves, such as shown on figures 9 and 10, would be obtained. These two locomotives, both of the New York Central Railroad, and tested under similar conditions, give maximum capacity figures, which, according to Mr. Wilson, differ from those obtained by the author by 17 % in one case and 42 % in the other. Similar relation of discrepancy would be obtained if the Cole, or the Vincent, or the Kiesel formula were applied, although the figures might be different. No method would satisfy both capacity curves, and therefore these high power figures must be partly ascribed to local conditions, such as quality of fuel and method of operation, which may be different on different roads, and partly to the design of the different parts, which cannot be expressed by a formula. It would not be advisable to have a yardstick formula for comparison based on the capacity of the 4-6-4 J1 locomotive. These capacity curves every railroad has to find out for itself.

As to the value of capacity tests, it has been also pointed out in the paper that unless there is complete assurance that the capacity tests have been made without exceeding the sustained evaporation of the boiler, the capacity figures are likely to be exaggerated and cannot be taken as basis of locomotive performance.

Mr. Ching Pong Pei raises several interesting points. He seems to be somewhat puzzled that no reference has been made in the paper to the rate of firing.

As far as Mr. Cole's figures are concerned, the author is to blame for not having stated definitely that Mr. Cole was considering the rate of firing of 120 lb. per sq. foot per hour. As the author made reference to Mr. Cole's publications and the « A. L. Co. Handbook », he did not think it necessary to repeat Mr. Cole's firing rate. Personally, he does not attach much significance to the rate of firing in the case of modern locomotives. When Mr. Cole devised his figures, the grate areas were comparatively small. Modern locomotives have much larger grate areas, and as stated in the

paper « in well-designed locomotives the proportions are such that the two factors more or less balance », the two factors being the limitation of the amount of coal which can be burned on a certain grate and the amount of heat which can be absorbed by a certain heating surface. In modern locomotives the second is more often than the first the ruling limitation, and it is therefore permissible to consider the evaporation as determined by the heating surface rather than by the grate area.

In view of the foregoing, smaller firing rates than what Mr. Cole figured on are now actually being used. In his paper the author was not interested in firing rates in view of the indefiniteness of the figures and also for the reason that they fluctuate too much. Moreover, they cannot serve as indications of forced or normal operation of a locomotive, as this depends upon the relative size of the grate. Therefore, the author would not be in favor of Mr. Pei's suggestion to plot a series of tractive-effort curves for various rates of firing. This would be a very indefinite basis for comparison, although it may be an interesting method of scientific investigation of locomotives of certain types.

The author already pointed out in the paper that, stricly speaking, tractive effort becomes a definite conception only when the conditions at which the tractive effort is produced are specified, as for instance the maximum tractive effort. the most economical tractive effort, the constant-evaporation tractive effort. Mr. Pei's suggestion is along these lines—he would have constant - firing - rate tractive efforts. Much more reliable conclusions can be drawn if these curves are plotted for different constant-evaporation rates. A still better way of comparing locomotives, at least for some purposes, would probably be on the basis of equal overall efficiencies.

However, the difficulty with all these methods, in which a system of curves is considered, is that during actual operation the locomotive is not burning a constant amount of coal per unity of grate area per hour, nor is it generating a constant amount of steam per unit of heating surface per hour, nor does it work at a constant efficiency. These factors are varying all the time, and in the opinion of the author, the higher rates of firing and evaporation are obtained at higher speeds. If anything, the locomotive is more likely to work at constant efficiency at certain modes of operation—performance, maximum performance, capacity. If a tractive-effort curve obtained from test performance, or capacity, be plotted over a chart consisting of a series of curves for different constant rates of firing or evaporation, probably it will be found that the actual tractive effort intersects all the other curves. If, therefore, a chart as suggested by Mr. Pei is presented to a railroad operator, as he recommends, it would be necessary to give him a table or chart, indicating the speeds at which the different rates are actually materialized. This would enable him to determine points, one on each of the curves, and plot the actual tractive-effort curve.

The author preferred, therefore, not to be bound by such purely theoretical considerations and to consider the curves which from actual tests are known to be obtainable in every day's service. These, called *performance curves,* can be plotted as shown in figure 16 of the paper for the New York Central 4-6-4 locomotive. These curves correspond to reasonable overall efficiencies (about 6 to 7 %) and to firing rates for modern locomotives up to 90 to 110 lb. per sq. foot of grate area per hour, as can be figured out from the total evaporation (E), average evaporation of coal (7 lb. of water per 1 lb. of coal), and grate area of any of the foregoing examples.

The author is not in agreement with the second and third premises, quoted by Mr. Pei, as those on which the correlation between boiler evaporation and rotative speed of the locomotive is based. Neither of the two premises has ever been mentioned by the author; as a matter of fact, the meaning of the second is not clear to him. The correlation referred to is for him a matter of observation of innumerable locomotive tests and conclusions drawn from stationary tests. As it has been stated elsewhere, the ultimate recommendations of the method do not depend

upon the correctness of the assumed correlation.

Mr. Muhlfeld takes issue with the author in considering the type « E » superheater as a part of the « modern locomotive », asserting that type « A » superheaters are able to give just as high superheats as the type « E ». In this latter respect Mr. Muhlfeld is quite right. In his conclusion the author stated also that « this [the use of the Cole method for all locomotives with type « A » superheaters] should not imply, however, that type « A » superheaters can never develop the horsepower recommended by the new method. Locomotives are known that have given very high performance figures with type « A » superheater on good coal.

It is obvious that the locomotive power depends upon the proper superheat, and that if type « A » superheater is so dimensioned that it can insure as high a temperature as type « E » superheater, the performance of a locomotive equipped with this type of superheater should be just as good as that with the type « E ». It has also been stated by the author that it would be better to refer the performances not to the type of the superheater, but to the temperature of superheat. However, it was pointed out that « it was not thought advisable to give constants for various superheats for the reason that before a locomotive is tested, its superheat is not known, and therefore these constants would not be helpful in calculating the horsepower of a locomotive beforehand ». It was also mentioned that it would be more logical to base such constants on the relation of the superheating surface to the evaporative heating surface. The author tried to find a law for such a dependence, but so far such comparisons did not give conclusive results.

There is one important thing which justifies considering the type « E » superheater a feature of the « modern locomotive », especially American; namely, it is difficult, even impossible, to build a type « A » superheater with a sufficiently large heating surface in a large locomotive boiler, while the type « E » superheater permits doing it. The ratio of the superheating surface to the evaporative heating surface in the largest type « A » superheater is only about 0.32, whereas it is possible to design a type « E » superheater with a ratio of 0.45. The majority of locomotives with type « E » superheaters has a ratio of 0.41 to 0.42. Mr. Muhlfeld cites certain examples in favor of the type « A » superheater which are small locomotives. Likewise, all European locomotives, which have mostly type « A » superheaters, are of comparatively small size. The difficulty occurs only when locomotive heating surfaces reach 4 000 to 4 500 sq. feet for which the type « E » superheater seems to be indispensable.

Mr. McBride is of the opinion that the increase in evaporation capacity of a boiler equipped with feedwater heater, found by the author to be 7 %, is underestimated, and that 15 % would be a more correct figure. He may be right in that the extreme 15 % figure calculated by Mr. McBride on the basis of heat saving is correct when everything is in first-class condition and the locomotive is working at the peak of its capacity. For average conditions 7 % is a more acceptable figure, although it may seem somewhat conservative.

Mr. McBride does not believe that road tests for which the author's figures were taken can be sufficiently accurate. But tests made on the Pennsylvania testing plant with Ils locomotive equipped with a feedwater heater of the open type showed a saving in heat fluctuating between 4.7 and 10.2 %, of which 7.45 is the average. Therefore, when the evaporation figures found by the author were about 7 % above the Cole evaporation figure, he thought the most simple way of introducing the feedwater heater into the ratios would be leaving the Cole figure unchanged and adding 7 % to that for locomotives equipped with feedwater heaters—this being in good agreement with stationary test results. The author believes that 7 % is a fair figure for average operating conditions, for the performance curve on the basis of available experimental data.

I do not quite agree with Mr. Oatley when he says that « the new method offers form-

ulas somewhat more complicated than have been used hitherto ». This would be true if we exclude the figure of boiler evaporation from the Cole method. However, ordinarily the Cole method requires the figuring of the boiler percentage, and in this case the knowledge of the boiler evaporation is also required.

Mr. Armstrong is right in calling attention to the discrepancies which might be found if the curves according to my method are compared with test results of locomotives equipped with improved drafting arrangements. Mr. Pownall also called attention to this fact, giving figures obtained from the Timken locomotive, which gave exceptional results at high speeds.

Table 15 is given by Mr. Sillcox in which new factors are suggested, differing from Cole figures by 9.2 to 9.5 %. They are marked « test »; it is claimed that they are based on recent tests and that the difference between them and Cole figures represents the advantage of the locomotive « constructed in 1933 as compared with 1910 », when M. Cole devised his method.

The author is not familiar with the tests to which Mr. Sillcox refers, and as no particulars of the tests or any data are given, the author is not in a position to dispute or confirm the suggestion of raising the Cole factors by 9.2 to 9.5 % for all speeds. He may only state this—when the need for a revision of Cole factors became apparent, the first thought was to increase Cole factors in a certain proportion. This was tried, but it was impossible to establish a uniform rate for all cases. This can be easily concluded from figures 9 to 14, in which the Cole curves, together with test performance curves, are shown. At low speeds (50 to 80, or 110 r.p.m. — piston speeds approximately 250 to 400, or 440 ft.p.m.) they were found in some cases above performance curves, and their further increase was not desirable (figs. 9, 11, and 12), while in others the whole curve lay below the performance curve (fig. 10), and still in others the curves touched each other at 50 r.p.m. The reason for this was evident—these locomotives had different boiler percentages. The next thing the author tried was to change the Cole curves in relation to the boiler percentages. He got more consistent results, but then it occurred to him that by doing so he eliminated the Cole values altogether, because the cylinder horsepower was thus introduced twice, in the numerator and the denominator, and these two values cancelled each other, leaving only a value proportional to the evaporation. Then the natural thing to do was to consider the evaporation only—what he did in his method.

The author explained in his paper the reasons for preferring to figure the locomotive characteristics on the basis of boiler dimensions rather than cylinder sizes. It is not necessary to repeat here the arguments, but it suffices to say that if we agree to limit ourselves to locomotives of a certain period and type, any essential part of a well-proportioned locomotive may be chosen as a yardstick for measuring its power. Superheating surface, weight, even length of a locomotive, could be selected for that purpose, and it would be found that the ratios of power to each of the enumerated dimensions fluctuate very little in modern locomotives. Nobody would seriously consider measuring the power of a locomotive by its length, but in the author's opinion, there is more justification in measuring it by the heating surface of the superheater (of a certain type) than by the size of cylinders. In table 22 horsepowers of all locomotives cited in the paper are referred to the product $pb \times A$ (boiler pressure times piston area), which represents the main factor in figuring the power of a locomotive according to Cole and also to Mr. Sillcox. The only difference between them is that in Cole's formula these products are multiplied by certain factors, while Mr. Sillcox would increase these factors by 9.2 to 9.5 %.

The ratios of column 3 of table 22 vary from 0.0230 to 0.0294, or 28 %. In the same table Cole evaporation figures Ec (not counting the increase due to feedwater heaters) and the ratios of power to these figures are also given for comparison.

TABLE 22.

Locomotive	Maximum I. H. P. (performance) 1	$pb \times A$ 2	Ratio of col. 1 to col. 2 3	Total evaporation, lb. per hr. 4	Ratio of col. 1 to col. 4 5
New York Central, 4-6-4	3250	110 447	0.0294	54 662	0.0594
New York Central, 4-8-2	3240	128 825	0.0252	59 514	0.0544
Lehigh Valley, 5100	3750	143 139	0.0262	70 530	0.0531
Lehigh Valley, 5200	3800	135 587	0.0271	71 694	0.0530
Timken, 1111	3650	143 139	0.0255	67 370	0.0542
Boston & Albany, A-1	3400	147 780	0.0230	62 958	0.0541

They fluctuate only 12.1 %, much less than ratios in column 3.

The dependence of Cole's method upon cylinder sizes will always make the correctness of his figures a function of boiler percentages. For switching and limited cut-off locomotives with comparatively small boilers and large cylinders, any revised Cole figures will be exaggerated, while for high-speed locomotives with small cylinders, such as in the New York Central 4-6-4 engines, the opposite will be the case. At the same time, the real source of power, the boiler, will not be taken into account. Mr. Sillcox's reasons for his recommendation to preserve the Cole method and modernize his figures are « familiarity, confidence, and universal acceptance » of the method. In the author's opinion, if it is agreed that the Cole method is not based on correct premises, the enumerated advantages of the Cole method are of little importance. As the art progresses, Cole figures will have to be revised from time to time, and while this in itself is not a handicap, the difficulty is that test results give no indication how to proceed with the revision of Cole factors. A summary rise of all, or even of several, factors will always be a very approximate solution of the question. Tests will always determine the two fundamentals: improvement (1) in boiler evaporation and (2) in steam consumption, irrespective of the boiler and engine design— whether Stephenson or water-tube firebox, high-pressure or low-pressure steam, simple-expansion or compound- or triple-expansion cylinders, or even a turbine instead of cylinders. As soon as these data are known, they can be immediately and directly applied to the suggested method and new constants determined, because the method is based on the same two fundamentals: boiler evaporation and steam consumption. It can not be done with the Cole method. This makes the method suggested in the paper flexible and really universal, although, maybe, not universal in Mr. Sillcox's sense.

[621.131.2]

Ratios of modern locomotives (1)

by H. S. VINCENT,
Formerly Chief Consulting Engineer, Franklin Supply Co.

(From *The Railway Mechanical Engineer.*)

Comments on the revised methods of easily applied coefficients for use with modern locomotives, proposed by A. I. Lipetz. A discussion of specific applications and comparisons with methods which are now being used.

* * *

The correct evaluation of the tractive force of a steam locomotive is important as it is the measure of its capacity to move traffic, and the yardstick by which the designer gages the succes of his effort.

Much time has been spent in evolving exact mathematical expressions based on elaborate analyses of the many variables which enter into a correct appraisement of locomotive tractive force; so far, none of these have been successful in accounting for the aberrations which occur in the locomotive cylinder. Designers have, therefore, been forced to rely on empirical methods based on preceding practice. This means of evaluating tractive force is still in effect.

In 1897, the American Railway Master Mechanics Association adopted a committee report fixing definite design ratios for steam locomotives, based on cylinder volume. These ratios largely governed locomotive design for the next 15 years.

The work of Dr. Goss, at Purdue University, and the elaborate tests of locomotives, conducted during the St. Louis Exposition in 1904, on the first real locomotive testing plant erected in this country, formed the basis for a more exact study of locomotive design. Of greater importance has been the pioneering work of the Pennsylvania Railroad in setting up its laboratory at Altoona, conducting exhaustive tests of locomotives and publishing the results for the benefit of other railroads and builders.

In 1914, the American Locomotive Company issued its bulletin entitled « Locomotive Ratios ». This was the work of Francis J. Cole, late chief consulting engineer for that company. The issue of this bulletin was epochal in locomotive design as it introduced the idea of rating the output of a locomotive boiler by evaluating separately the evaporation of its component parts. It completely displaced the older method of proportioning the boiler on the basis of cylinder volume. The flexibility of the Cole method permits its adaptation to radical changes in design of fire tube boilers. Consequently it applies as well today as when first promulgated.

Except for the introduction of the superheater and the normal increase in size and capacity, there has been less change in locomotive boiler design in the past 30 years than one might have expected. This is indicated by the design

(1) An abstract of a paper « Horsepower and Tractive Effort of Steam Locomotives (Locomotive Ratios) », presented at the 1932 annual meeting of the American Society of Mechanical Engineers, in New York, is reproduced on pp. 324-350 of this issue from *The Railway Mechanical Engineer*. The discussion of that paper and the author's closure omit Mr. Vincent's discussion. In this article Mr. Vincent has amplified his discussion and has prepared charts to illustrate many of his points.

TABLE I. — **Design ratios.**

	Ratio of evaporative heating surface to grate area.	Ratio of grate area in sq. feet to volume of two cylinders in cubic feet.	Ratio of evaporative heating surface in sq. feet to volume of two cylinders in cubic feet.
Ratios adopted by American Railway Master Mechanics Association in 1897, for bituminous coal	60	3	200
35 representative locomotive designs built 1914-1919 inclusive. All with two-wheel trailing trucks. 7—4-8-2; 8—4-6-2; 1—4-4-2; 10—2-10-2; 9—2-8-2.	58.7	3.56	207
39 representative locomotive designs built 1928-1932 inclusive. All with trailing trucks. 14—4-8-4; 4—4-6-4; 3—2-8-2; 7—4-8-2; 4—2-8-4; 1—4-6-2; 5—2-10-4; 1—2-10-2.	55.8	4.26	238
Of the 39 locomotive designs built 1928-1932, 27 with four-wheel trailing trucks	54.2	4.54	247.5
12 other designs with two-wheel trailing trucks .	59.7	3.65	217.5

ratios shown in table I. It is surprising to note the constancy of the ratio between heating surface and grate area, especially for locomotives with two-wheel trailing trucks, when one considers that at the time of the adoption of this ratio, there were no trailing trucks. The advent of the four-wheel trailing truck has caused the major change from the early design data.

In the bulletin « Locomotive Ratios », Mr. Cole gave « Speed Factor and Horsepower Curves » for determining locomotive tractive force and horsepower as related to piston speed. The curve indicating the ratios for saturated steam is identical with that used by the American Locomotive Company, for many years previous to the publication of the bulletin. Mr. Cole simply added another curve applying to locomotives using superheated steam. At the period when these superheated steam ratios were promulgated, there were comparatively few locomotives equipped with superheaters and these were of an early design giving relatively low superheat. It is, therefore, not surprising that the curve should fail to represent conditions which came about later, involving changes in superheaters, valve gears and cylinder design.

The most recent contribution to the subject of locomotive tractive force, is by Dr. A. I. Lipetz, consulting engineer for the American Locomotive Company, who presented a paper on the subject « Horsepower and Tractive Effort of Steam Locomotives (Locomotive Ratios) », before the Annual Meeting of the American Society of Mechanical Engineers in December, 1932. In this paper Dr. Lipetz discusses the various methods which have been used to evaluate locomotive tractive force and horsepower, including that of Cole, and finds that there is a discrepancy between the actual and calculated tractive force as determined by these ratios. Dr. Lipetz then proposes a new set of ratios or moduli based in part on the evaporation as determined from the Cole boiler constants, and in part on the steam rate at various speeds, developed in road tests of several modern locomotives. The proposed moduli are related to crank speed, rather than piston speed as has been the usual practice.

It will be recognized that the proposal

of Dr. Lipetz is the same in principle as that used for many years by locomotive builders but substitutes a new set of ratios assumed to express more suitably the tractive force and power of the present-day locomotive.

The Lipetz modulus *Mt* for tractive force is assumed to be constant for any given r. p. m. or piston speed, regardless of the maximum cut-off for which the locomotive is designed. Included among the six locomotives which are used by him as a basis for his steam consumption curve is one built with a maximum cut-off of 60 %, the others having relatively long, maximum cut-off. It is surprising to note from his figure 6 [1], that even at relatively low speed, the steam rate of the limited cut-off locomotive is shown greater than that of several of the long cut-off locomotives. This is so contrary to the results obtained in tests of a limited cut-off locomotive by the Pennsylvania and which can be easily demonstrated by analysis, that one is inclined to question the validity of that particular test. The answer may be found in the statement of Dr. Lipetz, that the tests which he used as a basis for his deductions, covered a limited range of speeds. It was therefore necessary for him to extend this range by a method which he explains in his paper.

From the writer's experience, there are but few road tests that can be used for establishing normal tractive force or horsepower constants, for the reason that the results obtained are average results, that is, the horsepower, the water and fuel consumption, the speed, etc., given as the average over a run, and unless one has access to the dynamometer record, or the test is made under constant conditions of speed and grade, the results are unsuited for establishing a rating curve. Road tests are, of course, invaluable to a railroad for giving information which can be used by the transportation department in fixing and adjusting tonnage.

Results obtained from operation of a locomotive on a test plant, such as that at Altoona, when intelligently analysed, give in the opinion of the writer, far more useful information for the construction of rating curves, than can be secured from road tests. In test plant operation, the uniform conditions under which each test is conducted, can be continued as long as is necessary, to secure the desired result. This is impossible in a road test. The apparatus for recording the data, can be operated more efficiently and accurately than is possible on a moving locomotive or car. The test is conducted by a staff, each of whom is thoroughly familiar with his duty.

In considering the subject of locomotive tractive force, it is necessary to define what is meant by the term. Tractive force is the product of the engine constant and the mean effective pressure developed in the cylinder. It is, of course, applied through suitable mechanism to the rail at the rim of the driving wheels but no allowance is made for frictional effects in transmission. There are at least three kinds of tractive force, viz. *Maximum tractive force,* or the effort delivered at any speed, without reference to the efficiency of boiler or cylinder and requiring a steam supply beyond the capacity of the boiler. This is *always greater* than the normal tractive effort. *Operating tractive force,* or the effort which a locomotive delivers in pulling a train, the tonnage of which has been fixed by the physical condition of the road and the necessities of transportation. This is *often less* than the normal tractive force. It is sometimes called performance tractive force. *Normal tractive force,* or the force delivered throughout the total range of linear speed of a locomotive designed and operated in accordance with the best known practice

(1) See page 328 of this issue.

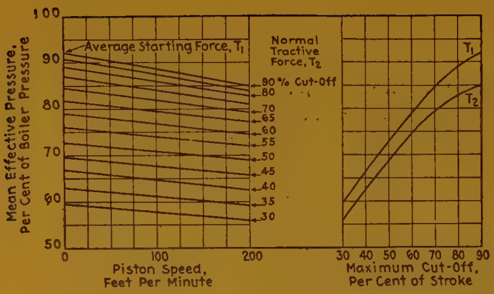

Fig. 1.

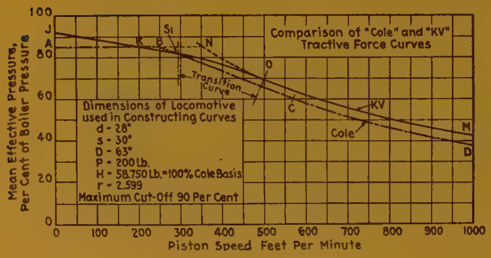

Fig. 2.

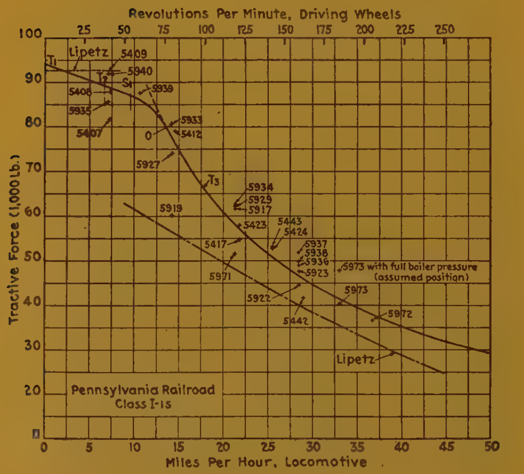

Fig. 3.

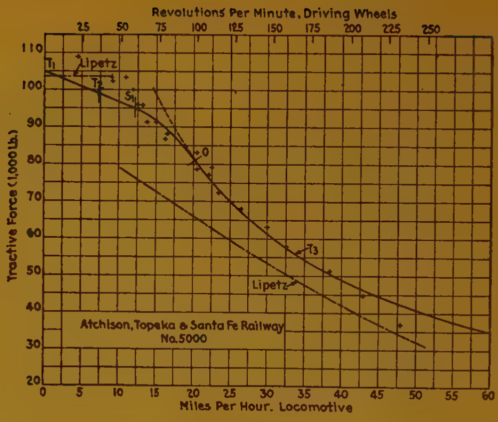

Fig. 4.

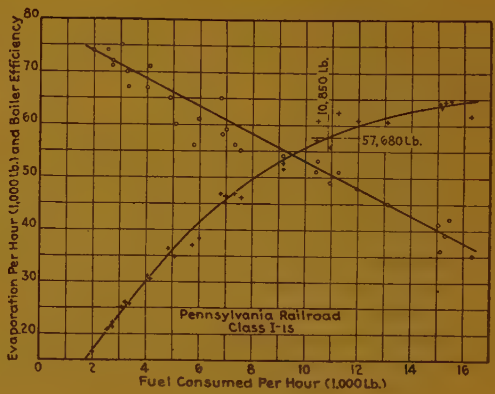

Fig. 5.

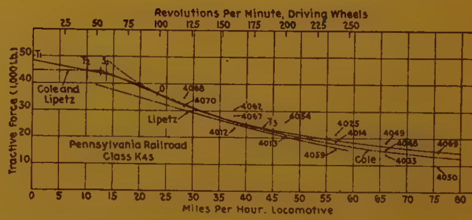

Fig. 6.

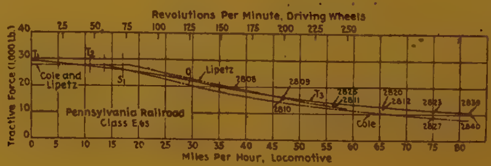

Fig. 7.

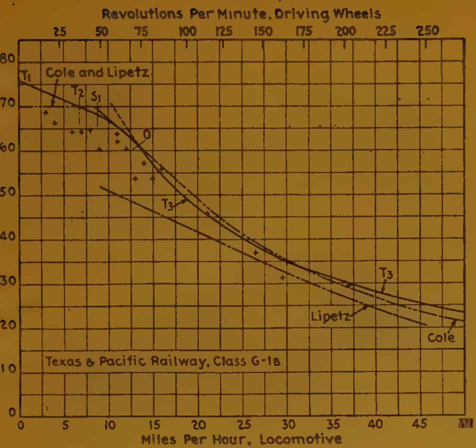

Fig. 8.

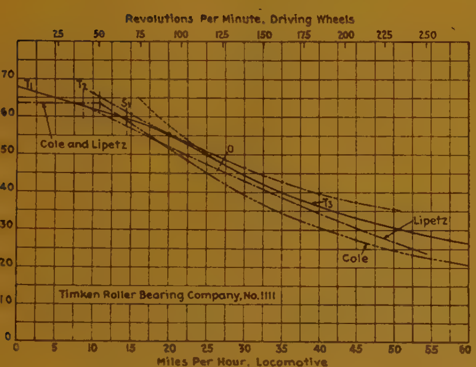

Fig. 9.

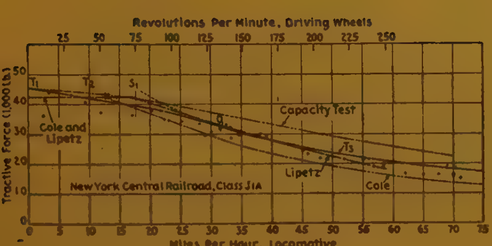

Fig. 10.

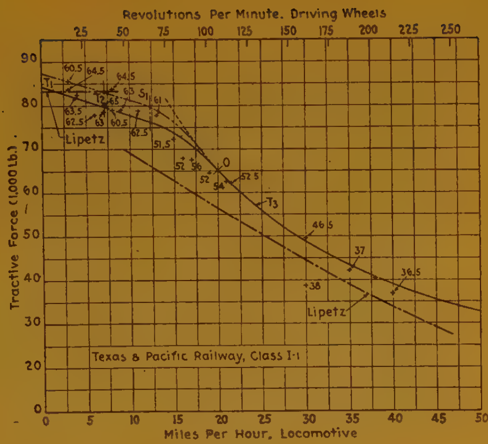

Fig. 11.

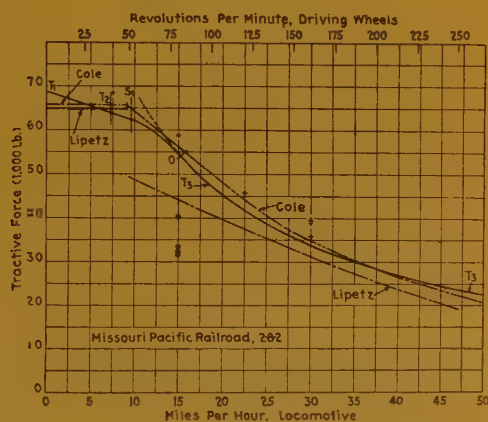

Fig. 12.

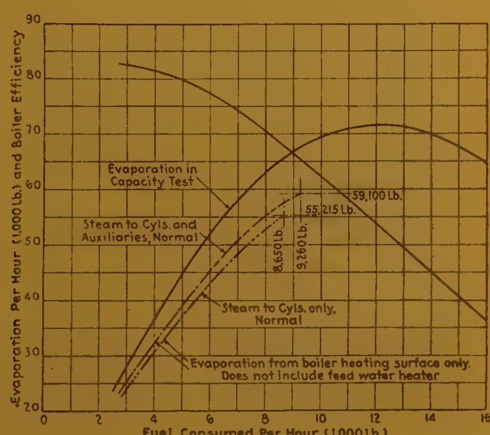

Fig. 13.

for its type and in which the fuel consumption and evaporation reaches but does not exceed a predetermined limit.

In addition, there is the *rated tractive force*, which is the measure by which the locomotive is usually compared with others of its type.

In establishing the rated tractive force of a locomotive, it is the custom to select some point on the speed-effort curve, calling this the nominal or rated effort of the locomotive. At this rating speed, the locomotive is assumed to develop in the cylinders a mean effective pressure of *aP*. From this is developed the well known tractive force formula

$$T = \frac{d^2saP}{D} \quad \ldots\ldots\ldots\ldots \quad (I)$$

in which

T = nominal or rated tractive force, lb.,
d = diameter of cylinder, inches,
s = stroke of cylinder, inches,
P = boiler pressure, lb. per sq. inch.
D = diameter of driving wheel, inches,
a = a factor depending upon the maximum cut-off.

TABLE II. — **M. E. P. in relation to cut-off.**

Maximum cut-off per cent of piston stroke.	Mean effective pressure in cylinder, per cent of boiler pressure.	
	Average at start or zero speed.	At piston speed of 200 feet per minute.
90	0.920	0.850
85	0.907	0.840
80	0.890	0.827
75	0.870	0.810
70	0.845	0.792
65	0.818	0.772
60	0.790	0.745
55	0.760	0.720
50	0.727	0.690
45	0.696	0.657
40	0.668	0.625
35	0.630	0.592
30	0.595	0.560

The value of *a* for various maximum cut-offs is given in figure 1, and table II. The broken lines in figure 2, indicate by *A-B-C-D* a graph based on the speed factors shown in table 14 of the *Locomotive Handbook*, American Locomotive Company, 1917, p. 24, generally known as the Cole factors for superheated steam. It is characteristic of these tables that a graph constructed from them, consists of a horizontal straight line *A-B*, a slanting straight line *B-C* and a hyperbola *C-D*, as indicated in figure 2.

The writer's experience has convinced him that the tractive force of a locomotive at low speed, cannot be indicated by a horizontal line such as *A-B*. It can be shown more accurately by the slanting line *J-K*, in which *K* represents the nominal or rated force and *J*, the average force at starting or zero speed. Many road tests which have been observed by the writer as well as the test records such as indicated in figures 3 and 4 would seem to substantiate this contention. Those who have ridden the footboard of a freight locomotive, know very well that a heavy train can be urged over the « sticking point » at a speed of 2 m. p. h. or less, when 8 or 10 m. p. h. would be impossible.

The vertical position and slope of the line *J-K*, depend entirely upon the maximum cut-off for which the locomotive is designed. It is evident that as the maximum cut-off is reduced the nominal or rated force for any given engine constant, must be represented by a lower percentage of the boiler pressure. For instance, as indicated in figure 1, a locomotive having a maximum cut-off of 90 %, has a rated force based on 0.85P,

whereas one having a maximum cut-off of 60 %, will be rated at 0.745P.

The position and slope of the line *J-K*, for maximum cut-off varying from 30 to 90 %, is given in figure 1. These data have been used successfully by the writer for several years, in representing the tractive force of modern locomotives. One very important result of adopting the slanting line *J-K*, rather than the horizontal line *A-B*, in representing tractive force at low speed, and of making *K* the rating point, is that it establishes a norm by which all locomotives of a given type may be measured and compared. The adoption of the horizontal line *A-B* permits any locomotive builder who might desire to do so, to claim a higher rated force for a locomotive than is justified by its dimensions, if a test shows that its tractive force comes at any point on the line *A-B*, although it is evident, that of two competing locomotives, that one is more powerful that gives its rating nearer *B*, than *A*.

There are six major factors involved in the calculation of the tractive force of a locomotive; these are :

a) the engine constant,

b) the normal flow of steam to the cylinders, lb. per hour,

c) the working pressure, lb. per sq. inch,

d) the maximum cut-off,

e) the average steam temperature in the cylinder,

f) the speed of the locomotive.

In the writer's opinion, the most practical formula for combining these elements in one equation to form the basis for a correct speed-effort curve, was devised by W. F. Kiesel, Jr., mechanical engineer of the Pennsylvania. The formula has been successfully used by Mr. Kiesel for many years, being derived from results obtained on the Altoona test plant, but modified to apply to road conditions. This formula, as slightly altered and transported by the writer, is

$$T_3 = \frac{1.95\, P_1 M}{1 + (36.66\, MS) \;/\; (Hv)} \quad . \quad . \quad . \quad . \quad (2)$$

in which

T_3 = indicated tractive force, lb.
P_1 = boiler pressure, less 10 lb.
M = the engine constant = d^2s/D.
S = linear speed of locomotive, m.p.h.
H = the normal steam production of the boiler available for the cylinders, in lb. steam per hour, as determined by the Cole boiler constants.
v = specific volume of steam at P_1 pressure and 100 degrees superheat.

For absolute accuracy *M*, in formula (2) and *d*, in formula (1), should be reduced in value to allow for piston rod and tail rod (where used); as a rule, this refinement is not considered necessary.

The complete derivation of the Kiesel formula as published in an article by the late A. I. Wood, in the *Railway Mechanical Engineer*, December, 1916, slightly modified by the writer, is here reproduced. In addition to the notation given elsewhere, let

E = a ratio, found by dividing the total volume of one cylinder, by the volume, which the steam passed through the cylinder in one stroke, would have at the initial pressure.
n = r.p.m. of driving wheels.
m.e.p. = mean effective pressure in cylinder.

then

$\frac{H}{60}$ steam per min., lb.

$$\frac{H}{60 \times 4 \times n} = \frac{H}{240n} = \text{steam per stroke}$$

$$\frac{\pi d^2 s}{1728 \times 4 \times v} = \text{weight of one cylinder-full of steam}$$

$$E = \frac{\pi d^2 s}{1728 \times 4 \times v} \times \frac{240n}{H}$$

$$\text{But } n = \frac{1056\,S}{\pi D}$$

$$\text{therefore } E = \frac{\pi d^2 s \times 240 \times 1056 S}{1728 \times 4 \times H \times \pi D \times v}$$

$$= \frac{d^2 s \times 36.66 \times S}{H \times D \times v}$$

$$\text{but } \frac{d^2 s}{D} = M$$

$$\text{therefore } E = \frac{36.66 MS}{Hv}$$

$$m.e.p. = \frac{1.95 P_1}{1 + E} = \frac{1.95 P_1}{1 + (36.66 MS)/(Hv)}$$

$$\text{since } T_3 = m.e.p. \frac{d^2 s}{D}$$

$$T_3 = \frac{1.95 P_1 M}{1 + (36.66 MS)/(Hv)} \quad \ldots\ldots \quad (2)$$

As that portion of the tractive force curve T_3-O, figure 4, is from formulations of the writer and the portion T_3 from a modification of the Kiesel formula, the entire curve will hereafter be designated as the KV curve.

It should be clearly noted that the term E, is not the usual expansion ratio, which is the total cylinder volume plus clearance, divided by the volume at cut-off, plus clearance. Except for the addition of the clearance volume, the term E would be identical with the expansion ratio, if the pressure at cut-off was equal to initial pressure.

It will be observed from the construction of formula (2), that in definite terms, it does not account for clearance volume, exhaust pressure, heat interchange or many other of the quantities usually considered in an exact mathematical expression. These factors are, however, all accounted for in the average mean effective pressure, which is the algebraic sum of all these entities and which is given by the expression $m.e.p. = (1.95\ P_1) / (1 + E)$.

TABLE III. — **Test data on Pennsylvania Railroad class I1s locomotive.**

1	2	3	4	5	6	7	8
					Mean effective pressure		
Test No.	Revs. per minute.	Steam per hour flowing to cylinders lb. = H_1	Actual cut-off per cent of stroke.	E	Developed in test	$\frac{1.95 P_1}{1+E}$	Lipetz modulus $\frac{H_1 Mt}{D} / \frac{d^2 s}{D}$
5926	80	30 464	31.6	4.035	98.6	93.0	60.1
5927	80	46 437	50.5	2.650	150.1	128.3	91.8
5933	80	51 918	57.0	2.367	163.1	139.2	102.5
5915	120	34 219	29.6	5.390	79.7	73.3	57.7
5917	120	56 338	49.3	3.275	124.7	109.5	95.0
5929	120	63 650	56.0	2.900	126.2	120.0	107.2
5921	160	37 739	31.0	6.518	68.3	62.3	52.8
5937	160	61 732	49.6	3.982	104.9	94.0	86.3
5938	160	63 192	51.3	3.892	101.0	95.8	88.2
5973	180	60 940	51.6	4.540	81.7	84.6	77.2
5972	200	54 680	39.9	5.625	74.0	70.7	62.6

In table III data are given from a laboratory test of the Pennsylvania locomotive class I1s (*Bulletin* No. 32). In some of the tests, there is considerable discrepancy between columns 6 and 7, but much less than that between columns 6 and 8.

For the purpose of showing the applicability of the KV formulas, a number of typical locomotive designs have been selected and their tractive force plotted, figures 3, 4, 6, 7 and 8, using this formula. Curves are also shown, based on the Lipetz moduli and the Cole ratios, where the latter are applicable. The locomotive designs selected are those for which the writer has sufficient data to indicate the actual tractive force devel-

oped either in a road or laboratory test covering a considerable range of speed. In these charts the tractive force is related to m.p.h., but in each case r.p.m. is also indicated.

The Pennsylvania class I1s, is represented by figure 3. The maximum cut-off of this locomotive is 55 % of the stroke. The crosses indicate the tractive force developed on test plant (*Bulletin* 32). Adjacent to them is the appropriate test number. A few positions are also plotted from an earlier test of a similar locomotive (*Bulletin* 31). Table IV gives pertinent data corresponding to the test numbers shown in figure 3. This locomotive has a ratio of grate area to the volume of two cylinders, of 2.59; consequently the unit fuel consumption shown in column 4, table IV, seems excessive,

TABLE IV. — **Test data on Pennsylvania Railroad class I1s locomotive.**

1	2	3	4	5	6	7	8	9
			Coal as fired					
Test	Revs. per min.	Indicated tractive force, lb.	per sq. foot of grate per hr., lb.	per hour, lb.	Steam per I.H.P.-hour, lb.	Equiv. fuel per sq. foot of normal grate, lb.	Boiler pressure, lb. per sq. in.	Boiler eff., per cent, incl. super-heater.
5933	80	80 600	132	9185	17.3	102.0	247.0	54.0
5412	80	76 900	140	9820	17.3	109.0	246.5	51.8
5927	80	74 200	98	6809	16.8	75.7	247.0	65.0
5919	80	60 350	70	4883	15.9	54.3	248.0	65.0
5917	120	61 600	150	10460	16.4	116.2	247.0	51.0
5423	120	56 400	166	11640	16.6	129.3	251.5	46.6
5417	120	53 400	202	14150	17.9	157.2	248.8	41.6
5971	120	51 600	101	7000	15.6	77.8	247.0	59.0
5424	140	51 400	157	10980	16.1	122.0	249.5	51.7
5923	160	47 700	189	13128	16.7	145.8	247.0	45.0
5922	160	44 600	132	9187	15.4	102.0	249.0	54.0
5442	160	40 750	119	8340	15.9	92.7	248.1	60.7
5973	180	40 350	235	16356	18.0	182.0	211.0	35.0
5972	200	36 560	158	10980	16.0	122.0	243.0	29.0

but reference to column 5, shows that the total hourly fuel consumption is normal. In figure 5, the evaporation and boiler efficiency of the I1s locomotive is given, as related to the total fuel consumption per hour. The normal evaporation of this boiler, available for the cylinders is 57 680 lb. steam per hour, as developed from the Cole boiler constants. It will be observed from figure 5, that this evaporation is reached at a fuel consumption of 10 850 lb. per hour, requiring a grate area on the Cole basis of 90 sq. feet. Column 7, table IV, indicates the unit fuel consumption which would have obtained had the grate area been made to conform to the Cole requirements.

The tests which most closely conform to the curve T_3, are 5933, 5412, 5927, 5423, 5424, 5923, 5922 and 5972. The equated fuel consumption for these particular tests, does not, except in one instance, greatly exceed the Cole limit of 120 lb. The boiler efficiencies as shown in column 9, are reasonably good for the existing conditions. Test 5973, which apparently comes near the curve T_3, would actually lie well above it if the boiler pressure were normal instead of being 211 lb. as indicated in column 8.

In figure 6, the curves T_1, T_2, T_3, indicate the tractive force of the Pennsylvania passenger locomotive class K4s (*Bulletin* 29) as plotted by the KV method. Curves are also shown based on the Lipetz moduli and the Cole ratios. As in the last example, the small crosses indicate the tractive force actually developed in the tests whose numbers are adjacent the crosses.

It will be observed that for this locomotive the KV and Cole curves are practically identical at speeds from 10 to 40 m.p.h. but diverge at the latter speed. This is to be expected as the K4s is one of the locomotives on which Mr. Cole based his ratios. The Lipetz curve also approximates the others at medium speeds.

Data from *Bulletin* 29 are shown in table V for the test numbers indicated in figure 6. The tests which most closely approximate the curve T , are 4070, 4014, and 4069. As indicated in column 4, of table V, the unit fuel consumption for these tests is well within the range of good practice. The curve T_3, lies about midway between the pairs of tests 4012 and 4067, also the pairs 4048 an 4049. It will be seen from a comparison of the unit fuel consumption for each of these tests, as indicated in column 4, table V,

TABLE V. — **Data on Pennsylvania Railroad class K4s passenger locomotives.**

1	2	3	4	5	6	7
			Coal as fired			
Test No.	Revs. per min.	Indicated tractive force, lb.	per sq. foot of grate per hr., lb.	per hour, lb.	Steam per I.H.P.-hour, lb.	Boiler eff., per cent., incl. super-heater.
4068	120	34 900	132.6	9180	21.2	57.1
4070	120	31 700	86.6	.6000	18.5	67.0
4062	160	30 350	167.5	11580	21.2	50.3
4067	160	28 150	101.9	7060	18.2	65.6
4012	160	23 480	71.3	4940	17.0	69.8
4054	200	25 950	173.3	12000	20.5	49.9
4013	200	20 980	90.9	6300	17.0	62.2
4025	240	21 380	139.4	9650	18.7	56.6
4014	240	18 620	92.9	6430	17.4	66.2
4059	240	15 980	68.3	4730	16.2	73.5
4049	280	18 370	145.6	10080	18.4	54.5
4048	280	15 690	83.2	5760	17.0	72.9
4033	280	13 690	63.4	4390	17.0	80.1
4069	320	14 920	126.5	8760	18.3	54.9
4050	320	9 510	49.2	3400	15.4	75.8

that the apparent consumption indicated by the position of curve T_3, is not excessive. It must not be inferred from this, that the fuel consumption is necessarily proportional to the position of these test points.

Referring again to figure 2, the sloping line *J-K*, is extended to S_1. At the latter point the straight line becomes tangent to the transition curve which connects the straight portion with the hyperbola. The locus of S_1, is a function of the boiler capacity. Its location may be determined graphically by drawing a horizontal line through *K*, intersecting the hyperbola at *N*; S_1 then, equals 0.84 *A-N*. The position N, is the point on the hyperbola where the cylinders would utilize the normal boiler capacity at maximum cut-off, if the tractive force remained constant from *A* to *N*. Stated as an equation,

$$S_1 = \frac{(1.95 P_1 M) - T_2}{(36.66 M T_2) / (0.84 H v)} \quad . \; . \; . \; . \; . \; . \; . \quad (3)$$

The transition curve between the straight portion of the graph *J-*S_1, and the hyperbola T_3, at *O*, is empirical.

An older design of passenger locomotive is represented by the Pennsylvania Class E6s (*Bulletin* 21); figure 7, indicates the tractive force of this locomotive as developed by the KV method. Curves are also shown constructed in accordance with the Lipetz moduli and the Cole ratios. The crosses indicate representa-

tive tests; data for which are given in table VI, figure 7, demonstrates that the actual tractive force as developed in tests 2808, 2810, 2825, 2812, 2823 and 2840 is represented very closely by the curve T_3. Reference to column 4, of table VI, will show that in none of these tests was the unit fuel consumption excessive as rated on the Cole basis. The boiler efficiencies column 7, are normal, considering the very short tubes with which this boiler is equipped (14 feet). The steam con-

TABLE VI. — **Test data on Pennsylvania Railroad class E6s locomotives.**

1	2	3	4	5	6	7
			Coal as fired			
Test No.	Revs. per min.	Indicated tractive effort, lb.	per sq. foot of grate per hr., lb.	per hour, lb.	Steam per I.H.P.-hour, lb.	Boiler eff., per cent, incl. superheater.
2808	160	19 250	98.7	5450	19.5	61.4
2809	200	16 550	126.7	7000	18.8	49.9
2810	200	16 370	129.6	7170	19.0	48.7
2811	240	12 840	82.6	4570	17.1	64.2
2825	240	13 975	128.6	7110	17.9	48.6
2812	280	12 110	98.6	5460	16.8	58.9
2820	280	13 280	108.6	6000	16.9	58.3
2823	320	10 860	126.7	7000	16.4	45.0
2827	320	8 930	98.0	5420	16.7	50.3
2839	360	10 740	144.8	8000	16.3	44.7
2840	360	10 710	117.7	6500	16.0	55.9

sumption column 6, for the tests in question, show a constantly decreasing range, as would be expected in normal operation.

The three examples cited represent locomotives of an older design, the date they were initially built ranging from 1910 to 1916. In figure 4, is plotted, the tractive force of one of the most modern designs, No. 5000, class 2104, built by the Baldwin Locomotive Works in the latter part of 1930, for the Atchison, Topeka & Santa Fe. Curves T_1, T_2 and T_3, are drawn in conformity to the KV method. A curve is also shown employing the Lipetz moduli. The small crosses indicate the tractive force as developed in a road test, results of which were published in the *Railway Mechanical Engineer*, December, 1931. There is a remarkable correspondence between the curves T_2, T_3 and the test data, at all speeds up to 43 m.p.h. The test at 48 m.p.h. lies nearer the Lipetz curve, but at all other speeds, that curve is far from representing the test. At low speed, it will be seen that the test positions lie above the curve T_1, T_2. This locomotive has a maximum cut-off of 60 % and is equipped with the Chapman-Lanning starting valves which have very large starting ports, as compared with other limited cut-off locomotives. The effect of these large ports is a virtual increase in the cut-off to approximately 67 % during the time that the ports are open.

Of all the modern designs, the Timken locomotive No. 1111 is perhaps the most widely known, as it has operated over a number of railroads and has received much favorable notice in the technical journals. The tractive force of this locomotive is indicated in figure 9 by the curves T_1, T_2 and T_3. The Lipetz moduli and the Cole ratios are also represented by appropriate curves. The tractive force as developed in a test of the locomotive on the Lehigh Valley is indicated by crosses, through which a smooth curve is drawn. The position of the test data would indicate that they represent a capacity test and as such would not be correctly represented by the normal tractive force curve. It is interesting to note that

the tractive force developed in the capacity test, approaches the KV curves at 20-25 m.p.h., but rises above it at an increasing rate at higher increments of speed. The writer has not sufficient data to discuss the reason for this rather abnormal development.

One of the locomotives used by Dr. Lipetz, as a basis for the formulation of his modulus *Mt*, was the 4-6-4 passenger locomotive class J1a, built by the American Locomotive Company for the New York Central. In figure 10, tractive force curves are shown as constructed by the KV method for this locomotive. Other curves are given, based on the Cole factors and the Lipetz moduli. The small crosses indicate the tractive force actually developed in a road test and are scaled from figure 15 [1], of the paper by Dr. Lipetz. The curve based on the modulus *Mt*, closely represents the mean of this test, as might be expected. It will also be observed that the KV curve very nearly approximates the higher range of tractive force developed in the test, up to a speed of about 60 m.p.h. This bears out the contention of the writer that a tractive force formula should indicate the best that a locomotive can do under any normal operating condition. At speeds above 60 m.p.h., the test positions lie between the KV and the Cole curves, but evidently they lie above an extrapolation of the Lipetz curve. The curve indicated by the broken line in figure 10, represents a capacity test made by the railroad engineers. It is interesting to note that at zero speed, the curve is identical with position T_1, of the KV curve, but rises above it progressively with increasing speed. This test indicates what may be accomplished by forcing the boiler beyond its capacity and probably by a manipulation of the water level, commonly known as « trading water for steam ».

In figure 8, data are given for a class 2-10-2 locomotive built in 1916 and in figure 11, for a class 2-10-4 locomotive built in 1925, both for the Texas & Pacific. In both cases tractive force curves in accordance with the KV method have been plotted. For the 2-10-2, curves are also plotted in conformity with the Cole ratios and the Lipetz moduli. For the 2-10-4, the Lipetz curve only is shown, as the Cole ratios do not apply to limited cut-off locomotives. The small crosses in both figures 8 and 11, represent the tractive force developed in road tests conducted by the engineers of the railroad. In figure 8, the test results at low speed, lie below the curve T_1, T_2. The writer does not have sufficient data to account for this discrepancy. At speeds above 10 m.p.h., either the T_3, or the Cole curve fairly represent the test results. In figure 11, the test positions lie above the curve T_1, T_2, at low speed. These locomotives have a nominal cut-off of 60 % but can be operated at 65 % and were so operated when the test was made. This would raise the curve T_1, T_2, as indicated by the broken line in figure 11. Curve T_3, closely conforms to the test positions. The cut-off at which these tests were made is indicated by figures adjacent to the crosses.

To prove that the KV method applies equally well to multi-cylinder locomotives, figure 12 is shown, representing a three-cylinder locomotive, 2-8-2 type, built by the American Locomotive Co. for the Missouri Pacific in 1925. Curves T_1, T_2 and T_3, are constructed on the KV basis, Cole and Lipetz curves are also shown. The small crosses in figure 12, indicate the result of a test of the locomotive made on the plant at Altoona and published in the *Railway Mechanical Engineer*, July, 1925, p. 462. One group of these are styled « capacity tests » and the other « efficiency tests ». The former are indicated in figure 12 by crosses, and the latter have the crosses enclosed by circles. The trend of the capacity test positions practically parallel the curve T_3, but lie above it for reasons previously discussed.

---

(1) See page 329 of this issue.

In figures 3-4 and 6-12, inclusive, the nominal or rated tractive force of the various locomotives is indicated, using the factors given by Dr. Lipetz, in the « author's closure » of his paper. Reference to figures 3, 4, 6, 8, 11 and 12, shows that the curve based on the modulus M*t*, at low speed does not connect with the straight line proposed by Dr. Lipetz for representing nominal or rated force. It would, therefore, be necessary to use some form of transition curve to connect them. Dr. Lipetz does not give any data for constructing such a curve.

The examples here cited, are sufficient to prove that the KV formulas very closely represent the normal tractive force of a steam locomotive. The writer has applied these formulas in plotting the tractive force of numerous locomotive designs and has never found one case where they do not apply.

It is undoubtedly true that in the majority of cases, a locomotive test discloses a drop in the evaporation curve as the higher speeds are reached, depending upon the characteristics of the locomotive. Mr. A. Giesl-Gieslingen in discussing the Lipetz paper, gave a curve of steam production and boiler efficiency as developed in the capacity test of the New York Central locomotive, class J1a. These curves are reproduced in figure 13, and related to the hourly fuel consumption. In this test, in which the boiler was pushed beyond its normal limit, the evaporation curve began to droop at a fuel consumption of 12 000 lb. per hour, equivalent to a unit consumption of 147 lb. per sq. foot of grate per hour. On the Cole basis of 3.25 lb. steam per I.H.P. hour, the normal hourly fuel consumption at the Cole evaporation rating of 55 215 lb. of steam is 8 650 lb. Allowing an additional evaporation of 7 % for auxiliaries, the hourly fuel consumption would normally be 9 260 lb. The curve of steam production, figure 13, discloses an evaporation of 67 500 lb. per hour at a fuel consumption of 9 260 lb. or an increase over normal of about 14 %. This is an indication that the water level was not maintained during the test, as all such capacity tests are not run continuously, but intermittently for the various speeds, until sufficient test data are secured for a range of speeds. The normal evaporation consistent with the fuel consumption, is indicated in figure 13.

At normal evaporation and fuel consumption, which is the basis for the KV curves, the boiler should be able to furnish a uniform weight of steam to the cylinders, regardless of speed, so long as the fire is maintained in proper condition and a uniform weight of fuel is fired. The requirements for this condition are : *a*) the boiler is properly proportioned, *b*) the fuel burned does not exceed the normal hourly rate, *c*) the locomotive has a well designed nozzle and drafting arrangement, *d*) the steam using apparatus viz. the cylinders, valves and valve gear are so designed and constructed, that they will supply to the drafting apparatus, equal weights of steam per unit of time. It is the requirement *d*) that usually fails, causing the evaporation curve to assume the characteristic droop.

In the test of locomotive class K4s (P. R. R. *Bulletin* 29), the maximum evaporation per hour at 240 r. p. m. was 61 670 lb. with a boiler efficiency of 56.6 %. In a later test of this locomotive, in which the only change was in the nozzle and front end arrangement the evaporation at 240 r.p.m. was raised to 72 620 lb., an increase of nearly 18 %, while the boiler efficiency only dropped to 52.5 %. This illustrates the effect of a properly designed front end, in increasing the capacity of a locomotive. The results achieved by the Timken locomotive, were due, in no small measure, to an excellent design of drafting arrangement.

The normal tractive force, as plotted by the KV method, is a criterion by which the success of any locomotive design may be judged. This is not true of

an operating or performance tractive force plot, predicated on an actual train operation, as it merely indicates what the locomotive did under existing conditions at the time of the run.

In further explanation of the salient features of the curves T_1, T_2, and T_3, let us first consider the straight inclined portion T_1-T_2, indicated by J-S_1, fig. 2. The boiler has no effect on this portion of the curve, so long as it furnishes steam at the nominal pressure. The cylinder and valve construction principally limit the tractive force which will be developed at this period, assuming that there is no change in cut-off. It has been demonstrated many times that with constant cut-off, the mean effective pressure will gradually decrease with increasing speed, and this approximately constant rate of decrease begins from the moment of starting and ends when limited by the boiler supply. It is therefore evident that a horizontal line, such as *A-B*, figure 2, can not correctly represent the tractive effort at this period. It has been argued that the horizontal line *A-B*, represents the normal limit of adhesion; if this is true of any particular design, it merely means that the locomotive will be slippery at starting, for, regardless of the normal limit of adhesion at some pre-

TABLE VII. — **Simplified method for calculating the hyperbola T_3.**

Based on dimensions of Timken No. 1111 locomotive, fig. 9.

		Constants	Maximum cut-off.
d = 27 inches	H = 72 540 lb.	1.95	85 per cent
s = 30 —	v = 2 111 cub. feet		
D = 73 —	P_1 = 240 lb.	36.66	

$$\frac{72\,540 \times 2\,111}{36.66} = 4\,175$$

$$\frac{27^2 \times 30}{73} = 299.5$$

$$1.95 \times 240 \times 4\,175 = 1\,953\,000$$

$$4\,175/299.5 = 13.95$$

M.p.h.	hyperbola T_3
10	1 953 000/23.95 = 81 600
15	28.95 = 67 500
20	33.95 = 57 550
25	38.95 = 50 200
30	43.95 = 44 450
35	48.95 = 39 900
40	53.95 = 36 200
45	58.95 = 33 150
50	63.95 = 30 550
55	68.95 = 28 350
60	73.95 = 26 400

$$T_1 = \frac{27^2 \times 30 \times 0.908 \times 250}{73} = 68\,000 \text{ lb.}$$

$$T_2 = \frac{27^2 \times 30 \times 0.84 \times 250}{73} = 62\,900 \text{ lb. at } 200 \text{ ps} = 8.68 \text{ m.p.h.}$$

$$S_1 = \frac{(1.95 \times 240 \times 299.5) - 62\,900}{(36.66 \times 299.5 \times 62\,900)/(0.84 \times 72\,540 \times 2\,111)} = 14.4 \text{ m.p.h.}$$

determined speed, the actual limit at the moment of starting will be as represented by *J*, figure 2. It will be understood that the position *J*, is a mean result of the various possible positions which the cranks may occupy at the moment of starting, and assumes that the resistance will be sufficient to develop the full starting effort.

In locomotives constructed with a limited maximum cut-off, starting ports are utilized to give a virtual cut-off of approximately 80 %, at the moment of starting. Usually these ports are so limited in area that their effect is not noticeable beyond a speed of one or two m.p.h. For such a locomotive, the average tractive force at starting, will be higher than the point *J*, figure 1, but will immediately drop to the line *J-K*, after a few revolutions of the driving wheels. In cases where starting ports having relatively large area are used, and these are not automatically closed immediately after starting, the actual effort will lie above the line *J-K*.

The hyperbolic portion of the tractive force curve T_3, is influenced largely by the boiler performance, but actually it is a boiler-cylinder curve. If it were only a boiler curve, as some assert, no change in the cylinder would have any influence on it, but we know from numerous tests, that a cylinder with a short cut-off will use less steam per I.H.P.-hour than will an equivalent cylinder having a longer cut-off. Consequently, with a given flow of steam from the boiler, the tractive force at any given speed will be greater for the locomotive having the more economical cylinder. The curve T_3, is predicated on a constant flow of steam from the boiler at all speeds above that limited by the cylinders only, as previously explained. However, in a case where the flow of steam under normal rate of firing is proved by test definitely to decline with speed, a reduced value of *H*, may be used in formula (2), but this will be an unusual condition.

TABLE VIII. — **Principal dimensions of locomotives.**

RAILROAD	Pennsylvania.	Pennsylvania.	Pennsylvania.	Atchison, Topeka & Santa Fe.	Timken locomot.	New York Central.	Texas & Pacific.	Texas & Pacific.	Missouri Pacific
Class or serial number	I-1s	K-4s	E-6s	5000	1111	J-1a	G-1b	I-1	3-cylinder
Wheel arrangement	2-10-0	4-6-2	4-4-2	2-10-4	4-8-4	4-6-4	2-10-2	2-10-4	2-8-2
Cylinder diameter, inches	30 1/2	27	22	30	27	25	28 1/8*	29	23
Cylinder stroke, inches	32	28	26	34	30	28	32	32	32 outside / 28 inside
Maximum cut-off, per cent of stroke	55	90	83	60	85	86	89	60	80
Wheel diameter, inches	60.2*	79.5*	79.2*	69	73	79	61.5*	63	63
Boiler working pressure, lb. per sq. inch	250	205	205	300	250	225	200	250	200
Weight in working order, lb.									
On driving wheels	352 500	202 880	141 000	348 200	264 000	182 000	264 545	300 000	244 500
Total locomotive	386 100	309 140	234 200	502 600	417 500	343 200	339 390	448 000	340 000
Heating surface, sq. feet									
Firebox inc. arch tubes and syphon	287	306.8	254.5	592.0(1)	483(1)	281.0(1)	275	473	349(1)
Tubes, water side	1276	2643.7	1738.0	751.6(1)	832(1)	800.4(1)	2503	1033	2214(1)
Flues, water side	3236	1090.7	695.0	4791.5(1)	3805(1)	3404.6(1)	1033	3607	1223(1)
Total evaporative surface	4799	4041.2	2687.5	6135.1(1)	5120(1)	4486.0(1)	3811	5113	3786(1)
Grate area, sq. feet	70	69.3	55.2	121.5	88.3	81.5	70	100	66.3
Cole evaporation, lb. per hour	57 680	51 110	39 570	82 770	67 800	55 215	48 475	67 135	50 550
Plus 7 per cent for feedwater heater	61 718	...	...	88 560	72 540	59 080	51 900	71 835	...
Tractive force shown in figure	3	6	7	4	9	10	8	11	12

(*) Actual dimensions at time of test. — (1) Heating surface taken from published data.

Formula (2), applies as well to limited cut-off locomotives as it does to those with long cut-off, for the reason that when the maximum cut-off is shortened, the cylinder is correspondingly increased in diameter. Consequently, the value of *M*, is influenced by the reduced cut-off. For multi-cylinder locomotives, it is only necessary to use the correct value of *M*, in formulas (2) and (3).

The curve T_3, being hyperbolic in construction, continues its hyperbolic character beyond the point where the tractive force begins to be limited by the cylinder, as indicated by the dotted portion beyond *O*, in figures 3, 4 and 6-12. It is therefore necessary to join the hyperbola to the straight portion of the plot, and a wholly empirical transition curve is used for this purpose, as indicated by S_1-*O*, figure 2. No error of any consequence is possible in drawing this curve, as it must be tangent with the straight line at S_1, and with T_3, at *O*. It would not actually represent test results in most cases, if the straight line *J-K*, was made to intersect the hyperbola T_3. The transition curve reflects that period in which the predominant influence of the cylinder is gradually yielding to the predominant influence of the boiler.

The quantity *H*, in formula (2), is not the total normal evaporation of the boiler, but is that weight of steam which the boiler can furnish to the cylinders in addition to the steam required for auxiliaries. No other interpretation is possible of the example shown by Mr. Cole on p. 10, of the *Bulletin* 1017, «Locomotive Ratios» (F. J. Cole, American Locomotive Company, 1914). It has been amply demonstrated in the locomotive tests at Altoona, that a boiler will furnish at least 7 % more steam than is required for the cylinders, without exceeding normal fuel consumption.

The formula (2) may seem, at first sight, somewhat complicated, but actually, only a few minutes are required to obtain the necessary loci for plotting the curve, assuming that a slide rule is used. In the writer's practice, a simplified form of calculation, as suggested by Mr. Kiesel, is used. An example of this is given in table 7, using the dimensions of the Timken No. 1111 as the basis of calculation. The tractive force curve resulting is given in figure 9. The principal dimensions of the locomotives used in plotting figures 3-12 are given in table 8.

No claim is made that the formula (2), is exact or that it includes as such, all of the complex series of events that take place in the locomotive cylinder. The formula is empirical, but « it is based on data scientifically utilized to develop an empirical formula ».

[625. 143.2 & 625 143.3]

The burden of defective rails,

by CECIL J. ALLEN, M. Inst. T.

(*The Railway Engineer.*)

Some observations which are of unusual interest to railway maintenance engineers, concerning the problem of the defective rail, occur in an article in the 12-8-33 issue of the American journal the *Railway Age*. The article has been written by Mr. C. W. Gennet, Jnr., Vice-President of the Sperry Rail Service Company of the U. S. A., to the results of whose track researches with Sperry detector car equipment various references have been made in past issues of *The Railway Engineer*. Mr. Gennet has found the basis for his deductions in the latest report of the Rail Committee of the American Railway Engineering Association, which covers rail failure statistics for the year 1931, with the further illumination of data furnished by the working of the Sperry cars in the same year. For the whole of the United States and Canada the total of transverse and compound fissured rails discovered in 1931 by Sperry-operated detector cars was 5 012, and by cars operated by the A. R. A., 421, while 4 795 rails with similar defects were discovered by surface-men, making a total for the year of 10 228. Assuming these rails to have averaged 100 lb. per yard in weight and 39 feet in length, approximately 6 000 tons of new rail, or 40 miles of track, were required in a single year for the replacement of rails with this type of defect alone.

The cost of rail defects.

But these formed a part only of the defective tonnage. The detector cars found a further 17 577 rails suffering from defects of other kinds, most of which required as prompt replacement as the fissured rails. Roughly, ten thousand further failures of different descriptions were reported, bringing the total number of defective rails removed in one year to the disquieting total of over 37 800. About 60 % of these rails were replaced before actual fracture had taken place. Further, the A. R. A. statistics cover only rails which have been in the road from one to five years, five years being regarded as the maximum period during which freedom from any defective condition can be claimed. As Mr. Gennet remarks, this arbitrary limitation is unfortunate, as many thousands of failures occur in later years of service, which might completely alter the deductions drawn from what has happened during the first five years of life. Further, such limitation makes no allowance for the relative severity of service in any particular route, nor for the probability of increased resistance and longer life in the case of the heavy rail sections recently introduced. But even on the five-year basis, 22 000 tons of rails, equal to 140 miles of track, were required in the North American continent in 1931 solely for the replacement of rails which had proved defective in service after five years of life or less. This, incidentally, was equal to 2 % of the total of all sections of 85 lb. per yard and upwards rolled in 1931, and to 6 % of the 1932 rolled tonnage, which owing to the depressed conditions, was very much less than previously. Not only so, but during the same year there were 251 railway accidents

in the United States and Canada for which defective rails were given as the primary cause, involving the death of 9 persons and injury to 126 others.

Grouping the failures.

It is next a matter of importance to divide up these failures into classes, and, so far as is possible to determine the causes of failure in each group. As Mr. Gennet points out, there is in this matter another serious shortcoming in the A. R. A. statistics, which, apart from special figures concerning transverse fissured rails, make no distinction between other types of defect, so throwing the fissure into a prominence which is not warranted by the percentage of failures from this cause. In the course of the Sperry detector car operation during 1931, the percentage of defective rails located which showed transverse or compound fissures was 24.1; horizontal split heads formed 36.1 % of the total; vertical split heads 32.4 %; and all other types of defective rails 7.4 %. Allowing for the likelihood that some of the compound fissures began their existence as horizontal splits, the tranverse fissured rails probably form about 22 % of the total, and head-failed rails about 70 %, while web and base failures accounted for roughly 8 %. Of all these causes of failure the transverse fissure is undoubtedly the most dangerous, for there is no external indication whatever of the presence of the defect, whereas the head-failed rail usually — though not always — gives ample warning of impending failure, either by the appearance of longitudinal cracks along or at the side of the head or in the fishing angle under the head, or by a spreading or bulging of the head under load. For this reason it is, perhaps fortunate that transverse fissures form less than one-quarter of the defects causing failure; but there are no grounds for other than concern over the figures which have been quoted.

Broken rails in Europe.

Some further light is thrown upon this problem by the statistics of rail failures which appeared in the September 1933 (*) issue of the *Bulletin* of the International Railway Congress Association. These figures presented, in a very detailed form, a survey of the rail fracture problem during the year 1931; each railway participating submitted a list of its failures, and divided them into classes, distinguishing between rails which, after breakage, showed transverse internal fissures; rails which broke through without revealing any kind of fissuring (probably failures due either to segregation, sections weakened by excessive wear, or fatigue caused by defective track maintenance); rails which after breakage showed longitudinal piping fissures in the web; and rails with defects which showed externally in the head or foot, accounting for breakage. Distinction was also drawn between failures at the rail-joints (in the fishing) and those in other parts of the rail; and between failures in tunnels and in the open respectively. These statistics are illuminating, though at the same time, for want of further information, inconclusive. It is unfortunate, too, that the participation of the railways was so incomplete. Particularly unfortunate, in view of what has been already mentioned in this article, is the entire absence of any American rail-failure statistics; in Great Britain the London & North Eastern Railway was the only company to supply figures; and none were forth-coming from Germany. But every French railway has participated, as well as the railways of Denmark, Italy, Rumania, Czechoslovakia, Japan, and certain railways in India and elsewhere.

From the figures of the railways whose

(*) French edition. — These statistics appeared in the December 1933, issue of the English edition.

information was the most complete, the two tables annexed have been compiled. The contrasts that they present are in some respects so extraordinary as to excite speculation as to whether the basis of failure-reporting is exactly the same in every case. On railways in Great Britain, for example, the fracture of a rail of « reduced section » — that is to say, a rail which has been planed to form a switch-tongue or the splice-rail or point-rail of a crossing — is not required by the Ministry of Transport to be reported in the same way as a broken rail whose section is reduced by head-wear, although defects may be clearly present in the planed rail. No such distinction appears justifiable. But in any event it seems clear from the statistics tabulated that running roads only are concerned — the French railways confine themselves to what are called « principal tracks » — and also that the figures relate only to definite fractures or ruptures, no rails being included which have been withdrawn prior to actual failure owing to the discovery of defects. Rails in which internal fissures had been detected by the use of the Sperry-car equipment, and which had been removed when the area of the fissure was within the danger limit, though no actual rupture had occurred, would in this way be ruled out, though the statistics were compiled in a year prior to the introduction of the first Sperry car into Europe. But in this respect also the statistics suffer from incompleteness.

The first table shows the total number of rail fractures during 1931 on 14 different railways, in relation to the single-track mileage comprised in the records and to the total train-mileage and ton-mileage of traffic passing over these lines in the year. This table furnishes one of the most remarkable tributes to British steel manufacture that has ever yet appeared in print. To every 1 000 km. of track the broken rails on the London & North Eastern Railway averaged 2.5 per annum; the next nearest figure to this is the 4.7 per annum of the North Western Railway of India, a large proportion of whose rails, supplied before the establishment of rolling-mills in India, are of British origin. In the French group the Paris-Orleans Railway has the best record, with 9.5 failures per 1 000 km. per annum, but the frequency on the remaining French lines ranges from 29.5 to 44.9 per annum. In Czechoslovakia it reaches 71.5 and in Rumania the very high figure of 97 fractures per 1 000 km. per annum — that is to say, practically one rail breakage a year in every 10 km. of line. In proportion to train-mileage and ton-mileage the British figures are just as noteworthy; furthermore, as seen in the second table, the British and Indian figures are achieved in spite of higher permissible axle-loads than those of any of the other railways with which comparison is made. Some explanation of the Rumanian and Czechoslovakian figures might have been sought in the high percentage of light rails used in those countries, but it will be noted in the second table that only one-fifth of the Czechoslovakian breakages were of the non-fissured type, which might be due to fatigue, or to a light section unduly stressed by excessive loading.

Fissures.

Transverse fissures account for roughly 10 % of the European failures; Great Britain was the only country with no such failures in 1931, whereas the Czechoslovakian percentages of transverse fissure breakages were as high as 20 % in light rails and 16 % in rails of medium section. Piping accounted for 16 % of all the failures of light rails an 26 % of medium-rail failures, while other defects in the head or foot, which might in some cases have been secondary piping spreading from the original nucleus in the base of the head out to the surface of the rail, were responsible for 32 % of the light-

Statistics of fractured rails, 1931.

COUNTRY and RAILWAY.	Single track mileage comprised in statistics.			Maximum permissible axle-loads (metric tons).		Total number of fractures.			Fractures in fishing: Percentage of total.			NATURE OF FRACTURES (Percentage of total fractures in each class of rail), Showing transverse internal fissures.			Showing no internal fissuring.			Showing piping in web longitudinally.			Showing external defects in head or foot.		
	Light.	Medium.	Heavy.	L	M. H.	L	M	H	L	M	H	L	M	H	L	M	H	L	M	H	L	M	H
Great Britain :	Km.	Km.	Km.																				
L. N. E. R.	2 692	14 441	...	20.3	22.9	0	42	...	0	10	...	0	0	...	0	67	...	0	2	...	0	31	...
India :																							
N. W. R.	7 815	4 596	...	17.3	22.9	33	25	...	0	0	...	19	28	...	21	52	...	3	0	...	57	20	...
France :																							
P. O.	5 775	4 959	108	19.0	19.0	38	57	8	39	42	66	13	9	0	29	18	0	11	30	75	47	43	25
State	7 418	5 535	87	20.0	20.0	324	55	3	28	67	100	11	7	0	43	9	34	19	51	33	27	33	33
Est	4 327	5 267	43	...	...	161	35	0	20	74	0	12	12	0	46	25	0	10	42	0	32	21	0
A. L.	2 262	1 549	12	18.6	18.6	90	36	0	69	81	0	5	6	0	26	28	0	40	44	0	29	22	0
Nord	1 625	4 746	8	18.8	18.8	151	93	0	27	77	0	13	5	0	45	19	0	9	39	0	33	37	0
P. L. M.	6 497	8 591	91	19.9	19.9	229	424	17	45	82	100	10	8	0	47	19	17	16	25	50	27	48	33
Midi	3 362	2 172	...	18.0	18.0	146	102	...	72	88	...	9	9	...	51	35	...	14	22	...	26	34	...
Holland :																							
Netherlands	No return given.			16.0	20.0	196	30	...	57	90	...	2	0	...	32	20	...	15	37	...	51	43	...
Denmark :																							
State	1 955	1 048	...	...	...	155	8	...	77	50	..	4	12	...	40	37	..	40	13	...	16	38	...
Czechoslovakia :																							
State	11 727	4 189	...	16.0	17.1	965	173	...	44	62	...	20	16	...	21	18	...	16	12	...	43	54	...

Note. — Light (L) = Laid with *Light* rails, under 42.5 kgr. per m. (85 lb. per yard).
Medium (M) — — *Medium* — 42.5 to 52.5 kgr. — (81-105 lb. per yard).
Heavy (H) = — *Heavy* — over 52.5 kgr — (105 lb. per yard).

Total rail fractures, 1931.

COUNTRY.	RAILWAY.	Per 1 000 km. of single track.	Per 10 000 000 train-km.	Per 1 milliard ton-km.	Total fractures.
		No.	No.	No.	No.
Great Britain	L. N. E. R.	2.5	2.5	2.2	42
India	N. W. R.	4.7	15.4	13.4	58
France	P. O.	9.5	16.0	3.1	103
Japan	State	28.5	26.7	7 5	499
France	State	29.5	52.0	15.0	382
"	Est	30.7	46.5	10.0	296
"	A. L.	33.0	42.0	10.0	126
"	Nord	38.2	40.0	9.0	244
Holland	Netherlands	42.2	41.0	—	226
France	P. L. M.	44.1	56.9	12.9	670
"	Midi	44.9	73.0	19.0	248
Denmark	State	54.3	67 0	35.0	163
Czechoslovakia	State	71 5	98.4	34 4	1 138
Rumania	State	97.0	213.0	258 0	1 111

rail and 35 % of the heavy-rail breakages. The ruptures which showed no sign of fissuring averaged 34 % in light rails and 29 % in medium rails. Fissures of one description and another thus definitely accounted for roughly one-third of the failures comprised within the table, and even though the non-fissure failures could be attributed entirely to service conditions — which is very unlikely — the rail manufacturers still would be responsible, by the defects in their product, for the incidence of at least two-thirds of the defects tabulated.

In the article in the *Railway Age*, to which attention was first drawn, Mr. Gennet goes on to make the claim that the primary cause of transverse fissures is still an unsolved problem, but the Sandberg researches into the subject, which have been described in these columns, make it impossible to agree with such a conclusion. It may be agreed that neither special conditions of track, nor of section, nor of rail promise immunity, and that a length of track which appears free from fissures at one examination may be found infested with them later on, because of the growth of the fissures from the nuclei contained in the new rail as it leaves the mill; and also that fissures are found in new rail as well as in old. But it cannot be agreed that no condition of the steel itself promises immunity. For the letter from Mr. H. Munro, of Sperry Rail Service in France, published in the September 1933 issue of *The Railway Engineer*, on which editorial comment was made, showed that complete immunity from fissuring had been found by the Sperry car operating in that country in the case of all rails over which it travelled that had been subjected to retarded cooling in the Sandberg oven. This goes to prove that the Sandberg researches have not only determined with accuracy that the first cause of fissuring is found in cooling conditions on the rail hotbanks at the mills, but that in their retarded cooling treatment they have also evolved an effective remedy. In the circumstances, and particularly in view of the facts revealed by the Sperry Company's own representative in France, it seems idle once again to reiterate the claim that « the primary cause of fissures is still an undetermined problem », as Mr. Gennet does in this article. So far as this country is concerned, the majority of British rail-mills have now installed the Sandberg oven, and apply the treatment as a matter of course to all

rails for home use, with the result that the transverse fissure is practically non-existent in new British rail. To the Cargo Fleet Iron Company, at whose Middlesbrough works, in collaboration with the patentees, the first experiments in retarded cooling were made, belongs the credit of this general application in Great Britain.

Piping.

Yet this still leaves the problem, less dangerous in its results, but far more widespread in its scope, of the headfailed rail. Split heads may arise from a variety of causes. There is primary piping, due to insufficient cropping of the ingot in course of rolling, which usually results in a vertical splitting, at the rail-end, from the web of the rail up to the head; but with the general practice in this country of planing both ends of every rail, detection during inspection of any but the most minute traces of primary piping is a comparatively simple matter, so that such rails can be intercepted at the source. Secondary piping, however, is, as indicated in the editorial article in the September issue to which reference has already been made a more serious question, as it is invisible in a new rail, and discloses itself in service only by the appearance of cracks or the bulging of the rail-head. Generally speaking, it may be traced to melting-shop methods, especially where high outputs are being sought from a limited number and capacity of furnaces. Teeming into the ingot-moulds and stripping are, in consequence, unduly accelerated, and the result is that, by contraction, cracks or cavities are formed in the interior of the ingot, under the pipe proper, and are rolled out into the finished rail as vertical cracks in the base of the head, often well clear of the rail-end. Secondary piping may be associated with a perfectly sound quality of steel, just as transverse fissures; though piping and segregation are often found together. Primary piping may be largely avoided by the cropping of a greater percentage of discard from the ingot at the mill; a cure for secondary piping is to cast the ingot with the wide end upwards, instead of with the small end upwards, as is now the practice with rail ingots at several British mills. But the first method reduces the yield of finished steel per cast, and the second increases the costs of production.

Enclosures in the steel.

From certain works another source of head failure in the past has been the practice of covering the bottoms of ingot moulds with loose wash-plates of mild steel, in order, as far as possible, to preserve the bottom of the mould from erosion as the hot metal is teemed in; but the risk of this practice is that the mild-steel plate will rise into the ingot, and form an enclosure of mild steel in the high carbon steel of the rail-head. In many cases this form of defect has caused a complete shattering of the rail-head after the rail has been in use for some time. Fortunately the practice of using these wash-plates has now been practically abandoned in this country. In any event it was confined to certain rail-mills only, and one of Mr. Gennet's contentions is thereby stressed — that if rail failure statistics are to be of the maximum value, they must not only be analysed out into all the various causes for failure, in so far as those causes can be determined with reasonable accuracy, but they must also be analysed according to the particular mills at which the rails have been made, with a view to preventing defect-producing tendencies in manufacture. Slag enclosures in the steel, due to carelessness in the melting-shop, can be another fruitful cause of failure.

Conclusions.

These are matters in which railways might do well to pool their information in regard to defective rails. Further, in

view of the high percentage of failures due to piping, it becomes a question whether the top rails from ingots, in which the vast majority of piping and associated defects are found, should not be specially marked, bought at a price lower than that of middle and bottom rails, and reserved, as far as possible, for secondary work. In any event it would seem desirable to day down in specifications that no rails required for switch and crossing work shall be cut from the top one-third of the ingot, suitable marking being adopted to make this distinction possible. It is undoubtedly a matter of congratulation to British steel manufacturers that, on the showing of these figures, their record is so remarkably clean in comparison with that of their Continental rivals; and some acknowledgment is also due to the competence of rail inspection methods in this country that so excellent a result has been achieved. Explanation is here forthcoming, in part at least, of the fact that for the past 25 years at least not a single accident involving loss of life — indeed, no serious derailment of any description — has been caused in the British Isles by a broken rail. Yet there is no justification for any slackening of effort towards securing a better quality of rail, even in this country. Manufacturers would do well to realise that the replacement by them without cost of a rail found defective does not altogether compensate the user, who at his own expense has to remove the defective rail from the road and replace it by another, probably with the accompaniment of badly delayed traffic until the replacement has been carried out. If a switch or crossing rail fails, the difficulty and cost of substitution is considerably greater, and to this must be added the wasted labour on preparing, planing, and fitting the rail for its switch or crossing service. As Mr. Gennet concludes his article, « new provisions should be made that will relieve the railroads more largely of the burden imposed by the cost of defective rails, caused by abnormal steel ».

[621. 132.8]

The "Franco" locomotive.

The conception of the « Franco » locomotive is based on the consideration that, if we are to have a sufficiently high tractive effort to haul the loads required by present traffic demands without having to fall back on double heading, we must have a greater number of driving or coupled axles than is available with the ordinary type of locomotive. The « Franco » locomotive provides an adhesive weight adequate for hauling the heaviest trains without exceeding the axle loads allowed by the permanent way and bridges.

As regards the steam generator, the present type of boiler has been retained on this locomotive : it must, however, be possible to fire the very powerful boiler provided. To increase the power and efficiency of the boiler a system of feed water heaters, making the fullest use of the heat in the combustion gases as well as in the exhaust steam, has been fitted.

Figures 1 and 2 show the general layout of the « Franco » locomotive. It consists of three distinct driving units, each carrying its own weight, connected by articulated couplings whereby the independence of the relative movements of each unit is assured. The wheel arrangement can be represented by C-1 + 1-B-1-B-1 + 1-C (or 6-2 + 2-4-2-4-2 + 2-6).

The frame of the centre driving unit carries the boiler. This boiler (fig. 3) has a common firebox for the two opposed barrels with their tubes. Each of the firebox sides has a fire hole and door near the barrel so that two firemen can work at the same time independently of each other. The firebox is divided longitudinally into two distinct parts by two longitudinal walls forming a water leg.

The end motor units each carry a feed water heater heated by the combustion gases; this heater consists of a cylindrical barrel with smoke tubes. In addition the lower part is fitted with a series of tubes through which part of the exhaust steam passes and raises the feed water temperature to about 100° C. (212° F.). The combustion gases leaving the boiler are taken to the feed water heater by articulated pipes and then pass through the smoke tubes in the heaters; this additional heating surface raises the temperature of the feed water close to the vaporisation temperature at the working pressure of the boiler. The feed water heaters are fed by two feed pumps drawing water from the tanks carried on the outer driving units; each pump is capable by itself of maintaining the water level in the boiler.

The blast is obtained in the usual way by that part of the exhaust steam not used in the feed water heaters.

Figures 1 to 4 show the 3 000-H.P. « Franco » goods locomotive built by the Tubize Locomotive Works of « Les Ateliers Métallurgiques » of Nivelles. It has been tested on the Luxemburg line of the Belgian National Railways Company and hauled, over long gradients of 16 m. per metre (1 in 63), a train of 1 214 t. (1 193 Engl. tons) at a speed of 24 km. (15 miles) an hour.

The leading dimensions of the locomotive are as follows :

Firebox heating surface	26.240 m² (282.4 sq. ft.).
Tube heating surface	225 m² (2 421 sq. ft.).
Total heating surface of the boiler. . . .	251.240 m² (2 703.6 sq. ft.).

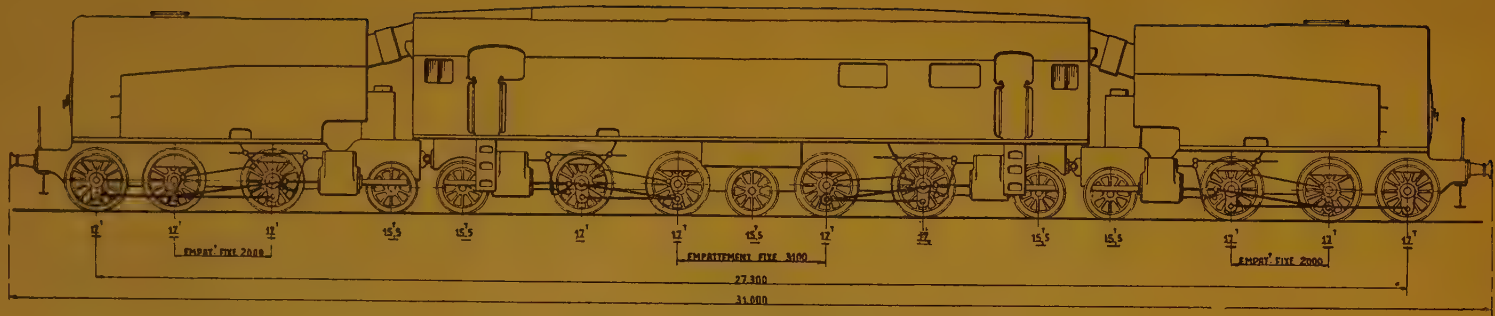

Fig. 1. — The « Franco » locomotive for goods trains.
Note : Empat. fixe = Rigid wheelbase.

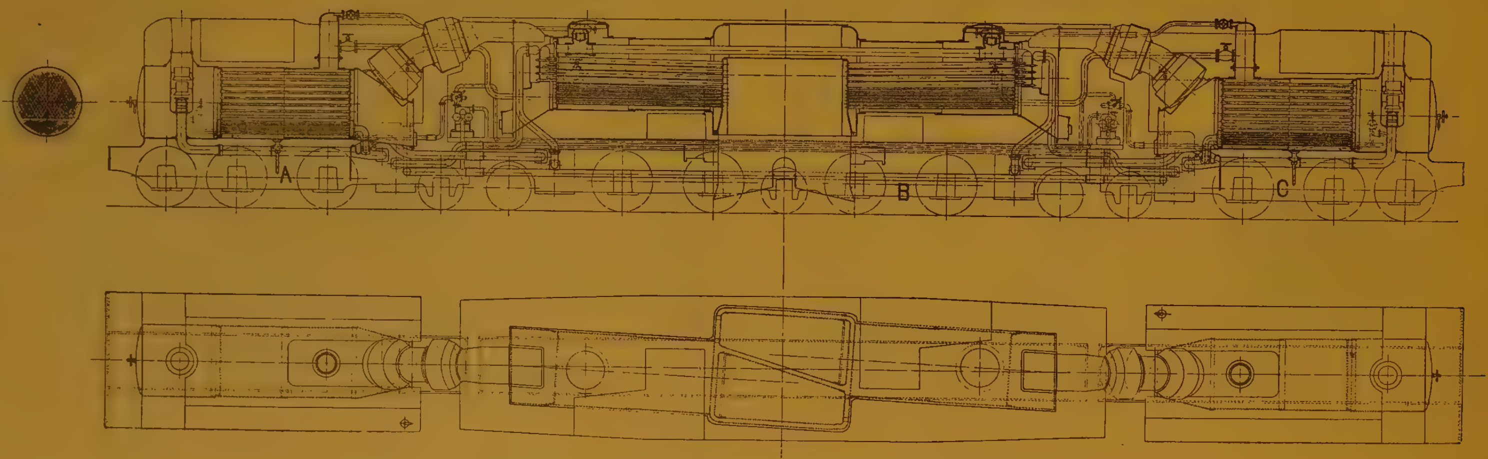

Fig. 2. — General lay-out of the « Franco » locomotive.

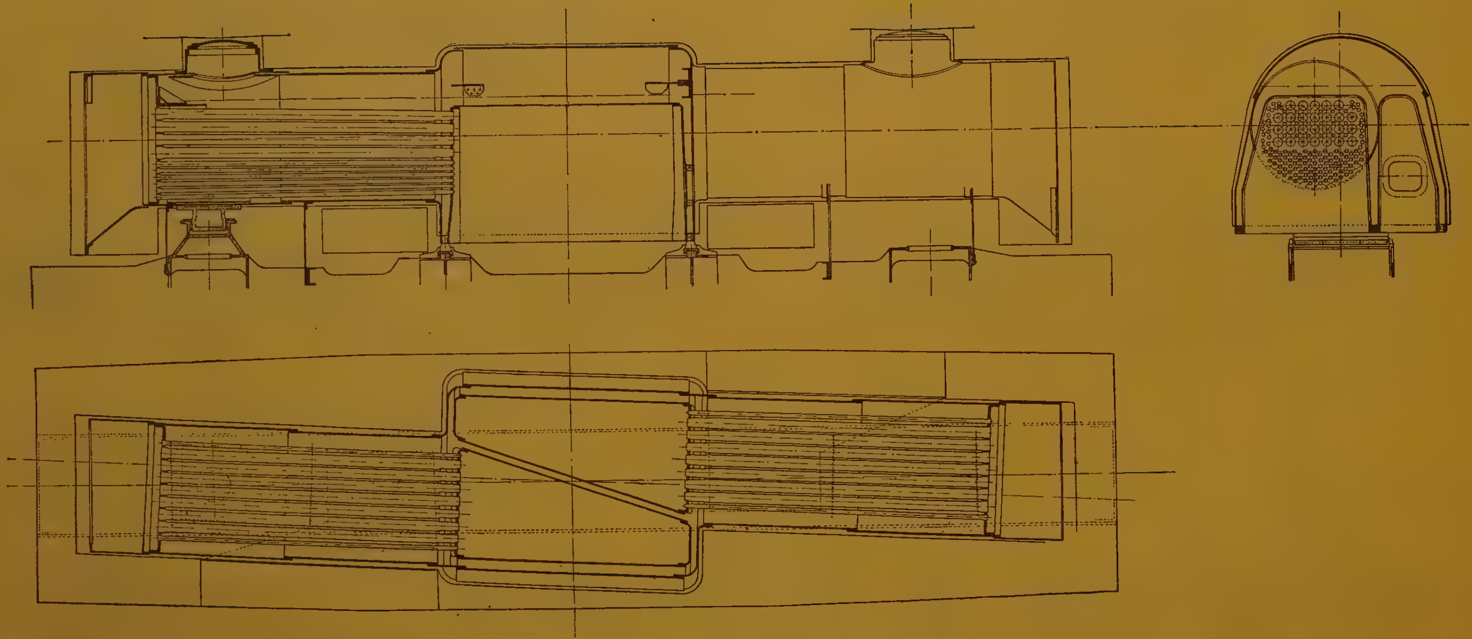

Fig. 3. — Arrangement of the « Franco » boiler.

Feed water heaters heating surface	268 m² (2 884 sq. ft).
Total heating surface	519 m² (5 585 sq. ft.).
Feed water heaters surface, heated by exhaust steam	43.290 m² (465.8 sq. ft.).
Superheater heating surface	77.220 m² (831 sq. ft.).
Grate area	6.250 m² (67.3 sq. ft.).
Number of cylinders	8
Cylinder diameter	435 mm. (17 1/8 in.).
Cylinder stroke	650 mm. (25 5/8 in.).
Diameter of driving wheels	1.370 m. (4 ft. 6 in.).
Diameter of carrying wheels	1 m. (3 ft. 3 3/8 in.).
Weight of the locomotive, empty	189 t. (186 Engl. tons).
Weight of the locomotive, in full working order.	248 t. (244 Engl. tons).
Adhesive weight	163 t. (160.4 Engl. tons).
Tractive effort $\frac{4 \times 0.75\ pd^2\ L}{D}$	37 500 kgr. (82 673 lb.).
Volume of water tanks	35 600 l. (7 830 Br. gall.).
Capacity of coal bunkers	9 tons.
Length of the locomotive, over buffers . . .	31 m. (101 ft. 8 1/2 in.).
Maximum speed per hour	60 km. (37.5 miles).
Minimum radius of curve round which locomotive can run	125 m. (6 1/4 chains).

The builders claim the following advantages for this type of locomotive :

1. *Fuel economy :* the temperature of the hot gases is brought down to 200 to 240° (392 to 464° F.); the feed water is raised in the heaters to some 150 to 180° C. (302 to 356° F.). The whole heating surface of the steam generators themselves is used to transform the water into steam, whereas in the boiler of the ordinary locomotive this surface has two functions.

At the same time as the temperature of the feed water is raised to that of evaporation in the feed water heaters, the scale forming salts in the water are precipitated in the bottom of the heaters so that the boiler is kept practically free from scale.

2. The heaters and the quantity of water they hold form a considerable and constant weight and this has made it possible to make the outer units driving units and increase the adhesive weight of the locomotive.

3. The speed of the gases through the smoke tubes of the heaters is less by about one half than that of the gases in the tubes of the boiler; as a result most of the sparks are deposited on leaving the boiler tubes and spark throwing is very much reduced.

4. The division of the firebox into two distinct combustion chambers and the use of two firemen makes it possible to clean one grate whilst the other is in use, an important advantage as regards evaporation.

5. With the « Franco » locomotive, very high superheat can be allowed as the heat units in the gases leaving the boiler tubes are recovered in the feed heaters. The engine is also very conveniently arranged for compounding; the four cylinders of the centre unit then become high pressure cylinders and the cylinders on the end units the low pressure. The articulated steam pipes between the units then contain low pressure steam only.

6. The « Franco » locomotive allows

Fig. 4. — The « Franco » locomotive during the trials.

the power and adhesive weight to be doubled without any alteration to the track. In addition the costs of maintaining the track are appreciably reduced by the fact that, at equal power :

a) the mechanism distributed over the whole length of the engine includes cylinders of small dimensions and the weight of the reciprocating parts is low;

b) the load on each pair of coupled wheels is lower;

c) the number of pairs of wheels coupled together in a group is much smaller.

7. The « Franco » locomotive is so arranged that it can be operated in both directions equally well.

8. Finally, the « Franco » locomotive is built of units similar to those of the ordinary locomotive : no new equipment is required in the building shops nor in the repair shops.

A. C.

[313 : 625. 143. 3]

Statistics of rail breakages for the year 1932.

(Concluded) (1).

We publish hereafter, in the form adopted at the Madrid Congress (1930) (2), the information supplied by member Administrations in connection with the rail fractures which occurred on their lines in 1932.

In the tables hereafter, and unless stated otherwise (3):

Light rails *applies to rails of a weight less than 85 lb. per yard (42.5 kgr. per metre),*

Medium rails, *to rails of 85 to 105 lb. per yard (42.5 to 52.5 kgr. per metre),*

Heavy rails, *to those weighing 106 lb. per yard (53 kgr. per metre) or over.*

---

(1) See first part in the *Bulletin of the Railway Congress*, February 1934, p. 176.
(2) See *Bulletin of the Railway Congress*, December 1930, pp. 2236, 2240-2242.
(3) See *Bulletin of the Railway Congress*, March 1926, p. 240.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
ALGERIA and TUNIS Algerian State Railways.		Miles			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
a) Algiers District																			
A. Rails outside tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	2	223.7	5.55	2	223.7	5.55	
A. Rails outside tunnels. *Medium.*	...	...	...	1	70.9	8.77	...	...	...	...	...	...	...	...	...	1	70.9	8.77	...
Total	...	...	...	1	70.9	8.77	...	...	...	...	...	...	2	223.7	5 55	3	291.6	6.32	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
B. Rails in tunnels. *Medium.*	...	...	...	...	...	...	...	...	...	3	111.8	16.66	...	...	...	3	111.8	16.66	...
Total	...	...	...	...	...	...	...	...	..	3	111.8	16.66	...	...	...	3	111.8	16 66	
C. The whole of A and B. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	2	223.7	5.55	2	223.7	5.55	
C. The whole of A and B. *Medium.*	...	...	...	1	70.9	8.77	...	...	...	3	111.8	16.66	...	...	...	4	182.7	13.60	
Total	...	...	...	1	70.9	8.77	...	...	...	3	111.8	16.66	2	223.7	5.55	6	406.4	9.18	

Number of train-miles: 2 388 955.

Number of English ton-miles: 260 594 300.

Number of fractures { total: 6.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 16.
per 1 billion tkm. or 612 000 000 English ton-miles: 14.03.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ≤ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D *Light rails.*	...	100 %	...	...	2	2	...
D *Medium rails.*	...	100 %	1	3	...	...	4
		Total	1	3	2	2	4
	Miles of single track of each class	*Light*	...	223.7		223.7	...
		Medium	70.9	70.9		...	145 4

		Light rails.	*Medium rails.*
E. *a)* New clean fractures	with internal transverse fissure	...	...
	without internal transverse fissure	...	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	1	2
	in the head	...	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	1	1

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
b) Oran District.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	43.9	...	...	106.6	...	...	104.4	...	...	58.4	...	26	507.7	31.82	26	821.0	19.678	10.4
B Rails in tunnels. *Light.*	...	...	...	...	0.1	...	...	0.1	...	...	...	...	...	0.14	...	...	0.34	...	...
C. The whole of A and B *Light.*	...	43.9	...	...	106.7	...	...	104.5	...	...	58.4	...	26	507.84	31.81	26	821.34	19.669	...

Number of { train-miles: 1 643 630. English ton-miles: 216 797 800.

Number of fractures { total: 26. per 10 000 000 tr.-km. or 6 250 000 train-miles: 98.29. per 1 billion tkm. or 612 000 000 English ton-miles: 73.24.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D *Light rails.*	3.846 %	96.153 %	26	...	...	6	20
		Total . . .	26	...	...	6	20
	Miles of single track of each class.		656.34	165.0		422.5	225.3

		Light rails.	
E. *a)* New clean fractures	with internal transverse fissure	7	26.923 %
	without internal transverse fissure	18	69. 23 %
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	1	3.846 %
	in the head	...	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	...	...

d) Number of pieces rails are broken into { 2 pieces: 23. 4 pieces: 1. 1 rail broken at the base of the web (21 in.). 1 rail with head come off on 4 ft. 3 in.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
c) Bône district		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	4	237.6	10.04	4	237.6	10.04	
A. Rails outside tunnels. *Medium.*	...	81.7	...	...	15.9	...	1	42.9	14.49	...	30.8	...	...	...	...	1	171.3	36.29	
Total	...	81.7	...	...	15.9	...	1	42.9	14.49	...	30.8	...	4	237.6	10 04	5	403.9	75.99	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	...	1.2	...	...	1.2	...	13.8
B. Rails in tunnels. *Medium.*	...	0.3	...	...	...	...	...	1.1	...	...	1.4	...	...	...	...	...	2.8	...	
Total	...	0.3	...	...	...	...	...	1.1	...	...	1 4	...	...	1.2	...	...	4 0	...	
C. The whole of A and B. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	4	238.8	10.14	4	238.8	10.14	
C. The whole of A and B. *Medium.*	...	82.0	...	...	15.9	...	1	44.0	...	...	32.2	...	...	...	...	1	174.1	35.70	
Total	...	82.0	...	...	15.9	...	1	44.0	...	...	32.2	...	4	238.8	10.14	5	412.9	75.26	

Number of train-miles: 1 120 970.

Number of English ton-miles: 326 600 600.

Number of fractures { total: 5.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 42.47.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ≤ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D *Light rails.*	...	10.44 %	1	2	...	...	1
D *Medium rails.*	...	36.29 %	1	...	...	...	...
		Total	2	2	...	7	1
		Miles of single track of each class.	223.8	128.0		296.8	115.0

E		*Light rails.*	*Medium rails.*
a) New clean fractures	with internal transverse fissure	...	...
	without internal transverse fissure	2	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	...	...
	in the head	1	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	1	...
d) Number of pieces rails are broken into		2	2

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
d) Constantine District		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		tons. English
A. Rails outside tunnels. *Light.*	...	83.9	...	...	0.6	...	...	19.3	...	...	106.9	...	...	366.6	9.23	9	577.3	9.23	
Medium.	...	8.1	...	...	0.6	...	...	15.5	...	...	...	...	...	...	...	...	24.2	...	
Heavy	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
Total	...	92.0	...	...	1.2	...	...	34.8	...	...	106.9	...	...	366.6	9.23	9	601.5	9.23	
B. Rails in tunnels. *Light.*	...	1.1	...	...	...	...	...	...	...	...	...	...	...	...	...	...	1.1	...	
Medium.	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	13.8
Heavy	...	...	...	...	...	...	...	...	...	...	...	...	...	2.2	...	...	2.2	...	
Total	...	1.1	...	...	...	...	...	...	...	...	...	...	...	2.2	...	...	3.3	...	
C. The whole of A and B. *Light*	...	85.0	...	...	0.6	...	...	19.3	...	...	106.9	...	...	366.6	9.23	9	578.4	9.23	
Medium	...	8.1	...	...	0.6	...	...	15.5	...	...	...	...	...	...	...	...	24.2	...	
Heavy	...	...	...	...	...	...	...	...	...	...	...	...	...	2.2	...	...	2.2	...	
Total	...	93.1	...	...	1.2	...	...	34.8	...	...	106.9	...	...	368.8	9.23	9	604.8	9.23	

Number of train-miles: 1 822 400.

Number of fractures { total: 9. per 10 000 000 tr.-km. or 6 250 000 train-miles: 30.

	Percentage of fractures in the part		NUMBER OF FRACTURES:				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails*	...	100 %	9	...	...	...	...
Medium rails	...	...	...	...	...	...	...
Heavy rails	...	...	...	...	...	...	...
Miles of single track of each class.			495.9	108.9		418.2	186.6

	Light rails.	*Medium rails.*	*Heavy rails.*
E. *a)* New clean fractures { with internal transverse fissure	2	...	...
a) New clean fractures { without internal transverse fissure	3	...	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head { in the foot	4	...	...
b) … { in the head	...	...	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head { in the web	...	...	...
d) Number of pieces rails are broken into	2	...	...

French Colonies and Protectorates

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
AFRICA. Colonial Railways in French West Africa. (A. O. F.)		Miles.			Miles.			Miles.			Miles.			Miles			Miles.		English tons.
a) Thiès Niger Railway.																			
A. O. F. 25 kgr. (50.4 lb.)	...	...	...	1	344.2	1.80	...	...	...	...	...	...	...	...	...	1	364.2	1.80	10.8
Standard 26 kgr. (52.4 lb.)	...	...	...	3	69.6	26.78	...	...	...	...	...	...	...	...	...	3	69.6	26.78	
Vignole 20 kgr. (40.3 lb.)	...	...	...	...	...	...	...	...	...	...	...	...	31	370 3	52.01	31	370.3	52.01	
Total . . .	...	...	...	4	413.8	6. 0	...	...	...	...	...	...	31	370.3	52.01	35	804.1	27. 7	...

Number of train-miles : 1 162 520.

Number of fractures { total : 35. per 10 000 000 tr.-km. or 6 250 000 train-miles : 180.

	Number of fractures in the part: covered by the fishplates	Number of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	on a rising or falling gradient: > 10 mm. per m. (1 in 100)
A. O. F. (50.4 lb.) . .	...	1	...	...		...	...
Standard (52.4 lb.) . .	...	3	...	...		...	...
Vignole (40.3 lb.) . . .	...	31	26	5		2	3

		A. O. F.	*Standard*	*Vignole.*
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	...	...	30
	in the head	1	3	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	...	...	1

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
b) Conakry-Niger Railway		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
Light rails 50.4 lb. per yard, 26 ft. 3 in long.	...	...	...	...	...	...	...	...	...	...	46	...	1	365.4	1.7	1	411.4	1.5	...
Standard. 60.5 lb., 39 ft. 4 in. long.	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...

Number of train-km. : 389 032.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	
c) Ivory Coast Railway.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
Vignole, 19 ft. 8 in. long	...	...	...	...	...	...	1	Over 10 years. 182.7	...	...	...	...	...	...	...	...	...	...	*Golwe engines 11.8 Engl. tons* (on heaviest axle when fully loaded: 13.7 Engl. tons).
Vignole, 26 ft. 3 in. long	...	...	...	1	18.6	...	...	13.7	...	...	...	...	...	...	...	...	...	...	
Vignole, 32 ft. 9 in. long	...	...	...	...	1.9	...	...	...	...	...	...	...	...	...	...	...	...	...	
Standard, 26 ft. 3 in. long	...	...	...	...	11.2	...	...	...	...	...	...	...	...	...	...	...	...	...	
Standard, 39 ft. 4 1/2 in. long	...	17.4	...	...	57.1	...	...	...	...	...	...	...	...	...	...	...	...	...	
Standard*, 39 ft. 4 1/2 in. long	...	108.1	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
Weight of rails: Vignole, 25.5 kgr. (51.4 lb. per yard). Standard, 26 1 kgr. (52.6 lb. per yard). Standard *: 30 kgr. (60.5 lb. per yard).																			
Total	...	125.5	...	1	88.8	7	1	196.4	3.16	...	...	...	...	...	...	...	...	...	

Number of train-miles: 555 832.

Number of factures { total: 2. per 10 000 000 tr.-km. or 6 250 000 train-miles: 22 5.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
INDO-CHINA Indo-Chinese Colonial Railways *a)* Northern System (*)		Miles.			Miles.			Miles.			Miles			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	...	...	1	185.8	3.35	...	...	...	...	...	...	12	381.5	19.54	13	567.3	13.39	10.4
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	2	1.9	666.67	2	1.9	666.67	
C. The whole of A and B. *Light*	...	...	...	1	185.8	3.35	...	...	...	...	...	...	14	383.4	22.69	15	569.2	15.40	

Number of train-miles: 1 401 948.

Number of English ton-miles: 187 301 430.

Number of fractures. total: 15
per 10 000 000 tr.-km. or 6 250 000 train-miles: 66.48.
per 1 billion tkm. or 612 000 000 English ton-miles: 48.98.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ≤ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails.*	...	100 %	10	4	1	12	3
	Miles of single track of each class.		509.5	95.7		575.4	29.8

		Light rails.
E. *a)* New clean fractures	with internal transverse fissure	4
	without internal transverse fissure	5
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	5
	in the head	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	1
d) Number of pieces rails are broken into	cracked	7
	2 pieces	7

Administrations and description of rails.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Maximum axle load
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
b) **Southern System** (*).		Miles			Miles.			Miles.			Miles.			Miles.			Miles		English tons.
A. Rails outside tunnels. *Light*	...	...	...	...	...	...	...	...	...	...	...	...	...	43.5	...	...	43.5	...	9.8
Medium	...	33.7	...	...	12.9	. .	...	...	...	...	25 2	...	2	246.2	5	2	318.0	3.9	
Total . .	...	33.7	...	...	12.9	...	...	...	...	...	25 2	...	2	289.7	5	2	361.5	3.9	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	..	...	...	..	...	...	...	...	...	...	...	...	9 8
Medium.	...	...	...	...	0.14	...	...	...	...	...	...	...	...	...	...	...	0.14	...	
Total . .	...	...	...	...	0.14	...	...	...	..	...	...	...	...	...	...	...	0.14	...	
C. The whole of A and B *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	...	43.5	...	...	43.5	...	...
Medium.	...	33.7	...	...	13 04	...	...	...	...	...	25.2	...	2	246.2	5	2	318.14	3.9	...
Total . .	...	33.7	...	...	13.04	...	...	...	...	...	25.2	...	2	289.7	5	2	361.64	3.9	

Number of train-miles : 730 000.

Number of English ton-miles : 76 773 220.

Number of fractures { Total : 2. per 10 000 000 tr.-km. or 6 250 000 train-miles : 17. per 1 billion tkm. or 612 000 000 English ton-miles : 15.9.

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails* . . .	...	...	...	...	...	...	...
Medium rails . .	...	100 %	...	...	...	1	1
Total . . .			...	...	...	1	1

		Light rails.	*Medium rails*
E. *a)* New clean fractures	with internal transverse fissure	...	1
	without internal transverse fissure	...	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	...	...
	in the head	...	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	...	...
d) Number of pieces rails are broken into		...	...

* Saigon-Mytho line.
Saigon-Nhatrang line and branches.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			The whole of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
Compagnie française des Chemins de fer de l'Indochine et du Yunnan.																			
Light rails.	...	11.0	...	...	14.4	...	...	...	...	...	...	...	6 broken ; 16 cracked.	505.1	7 broken ; 19 cracked.	...	...	...	9.8

Number of train-km. : 1 417 515.

Number of fractures { 6 rails broken. 16 rails cracked.

Number of fractures per 10 000 000 tr.-km. or 6 250 000 train-miles : { 26 rails broken. 70 rails cracked

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			[illegible]			
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
GREAT BRITAIN London and North Eastern Railway.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles		English tons.
A. Rails outside tunnels. *Light.*	...	12.34	...	...	33.62	...	...	25.25	...	...	50.41	...	2	1 539.99	0.81	2	1 661.61	0.75	20
Medium.	3	1 367.99	1.37	12	1 404.34	5.31	3	1 235.76	1.52	1	1 365.74	0.46	16	3 488.61	2.87	35	8 862.44	2.47	22 1/2
Total . .	3	1 380.33	1.36	12	1 437.96	5.22	3	1 261.01	1.49	1	1 416.15	0.44	18	5 028.60	2.24	37	10 524.05	2.20	...
B. Rails in tunnels. *Light.*	...	...	...	...	0.44	...	...	...	...	...	...	...	...	2.04	...	...	2.48	...	20
Medium.	5	59.01	52.95	1	26.91	23.22	1	12.21	51.18	..	2.64	...	...	2.69	...	7	103.46	42.29	22 1/2
Total . . .	5	59.01	52.95	1	27.35	22.90	1	12.21	51.18	...	2.64	...	...	4.73	...	7	105.94	41.30	...
C. The whole of A and B *Light* .	...	12.34	...	...	34.06	...	...	25.25	...	...	50.41	...	2	1 542.03	0.81	2	1 664.09	0.75	20
Medium .	8	1 427.00	3.50	13	1 431.25	5.68	4	1 247.97	2.00	1	1 368.38	0.46	16	3 491.30	2.86	42	8 965.90	2.93	22 1/2
Total . . .	8	1 439.34	3.48	13	1 465.31	5.54	4	1 273.22	1.96	1	1 418.79	0.46	18	5 033.33	2.24	44	10 629.99	2.59	...

Number of train-miles: 101 461 864.

Number of English ton-miles: 10 895 464 218.

Number of fractures { per 10 000 000 tr.-km. or 6 250 000 train-miles: 2.71.
per 1 billion tkm. or 612 000 000 English ton-miles: 2.47.

	Percentage of fractures in the part		NUMBER OF FRACTURES:				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
				Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . .	..	100 %	2	...	...	1	1
Medium rails . .	21.43 %	78.57 %	37	2	3	10	23
		Total . . .	39	2	3	11	24
	Miles of single track of each class.		9 456.71	1 173.02		3 301.95	5 466.95

		Light rails.	*Medium rails.*
E a) New clean fractures	without internal transverse fissure	...	3
	with internal transverse fissure	2	28
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	...	...
	in the head	...	2
	both head & foo	...	7
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	...	2
d) Number of pieces rails are broken into		2	2 generally

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
INDIA, DOMINIONS, PROTECTORATES & COLONIES. AFRICA. Sudan Railways (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
Light rails. 50 lb.	...	108.7	...	...	362.9	...	...	...	...	...	13.0	...	5	807.8	3 84	5	1 292.4	2.40	12.2
Light rails. 75 lb.	...	0.6	...	...	38.5	...	...	38.5	...	...	37.9	...	3	559.2	3.33	3	674.7	2.76	15.8
Total	...	109.3	...	...	401.4	...	...	38.5	...	...	50.9	...	8	1 367.0	3.63	8	1 967.1	2.52	

Number of train-miles: 1 380 870.

Number of English ton-miles: 642 663 550.

Number of fractures { total : 8.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 35.99.
per 1 billion tkm. or 612 000 000 English ton-miles: 7.61.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *50-lb. rails*	40 %.	60 %.	5	...	...	5	...
D. *75-lb. rails*	...	100 %	3	...	...	3	...
Total			8	...	...	8	...

E.	*75.-lb. rails.*	*50.-lb. rails.*
a) New clean fractures — with internal transverse fissure	...	1
a) New clean fractures — without internal transverse fissure	2	1
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head — in the foot	1	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head — in the head	...	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head — in the web	2	...
d) Crushed or split head	...	1

* Sidings excluded.

OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
INDIA. Bombay, Baroda and Central India Railway (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
I. — Broad gauge.																			
A. Rails outside tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	4	308.139	8.113	4	308.139	8.113	5.5 15.6 10.0
Medium.	...	...	...	...	...	...	...	...	...	1	1 248.375	0.501	4	1 248.375	2.002	5	1 248.375	2.503	17 75 19.7
Total	...	...	...	...	...	...	...	...	...	1	1 248.375	0.501	8	1 556.514	3.2	9	1 556 514	3.592	...
II. — Metre gauge																			
Light rails (**).																			
41 1/4 lb. per yard	...	...	...	...	...	...	...	...	...	...	...	...	4	1 125.48	2.22	4	1 125.48	2.22	8.0
50 lb. per yard	...	...	...	...	...	...	...	...	...	...	...	...	39	1 314.52	18.54	39	1 314.52	18.54	10.0
60 lb. per yard	...	...	...	...	...	...	...	...	...	...	...	...	...	466.84	...	...	466.84	...	12.0
Total	...	...	...	...	...	...	...	...	...	...	...	...	43	2 906.84	9.191	43	2 906.84	9.191	...

Number of train-miles { broad gauge : 7 141 856. metre gauge : 8 383 000.

Number of English ton-miles { broad gauge : 3 413 299 002.

Number of fractures { total : B. G., 9; M. G., 43. per 10 000 000 tr.-km. or 6 250 000 train-miles : B. G., 7.88; M.G ., 32.06. per 1 billion tkm. or 612 000 000 English ton-miles : B. G., 1.61.

	Percentage of fractures in the part		NUMBER OF FRACTURES :				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
				Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. Broad gauge *Light.*	84 %	100 %	100 %	...	...	...	...
Medium.	...	100 %	100 %	...	...	...	...

		Broad gauge lines.	
		Light rails.	*Medium rails.*
E. a) New clean fractures	with internal transverse fissure	...	...
	without internal transverse fissure	...	3
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	4	2
	in the head	...	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	...	...
d) Number of pieces rails are broken into		2	

(**) **Metre-gauge lines.** (*Light rails.*)

All fractures except 4 were outside the portion covered by the fishplates. Percentage of breakages according to appearance of fractures is not available.

(*) B. G. = Broad gauge. — M. G. = Metre gauge.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Rails in use for Less than 5 years. Number of fractures.	5 to 10 years. Number of fractures.	10 to 20 years. Number of fractures.	20 to 30 years. Number of fractures.	More than 30 years. Number of fractures.	TOTAL.	Approximate length of the lines considered as single track.	Number of fractures per 1 000 km. or 625 miles	*Maximum axle load in service.*
1	2	3	4	5	6	7	8	9	10
ITALY. State Railways. (*)							Miles.		English tons.
Light rails	...	3	10	22	320 (**)	355	7 539.2 of which 249.2 in tunnels	29.2	16.2
Medium rails:									
In tunnels	...	16	109	6	...	131	362.3	224.7 } 22.5	19.7
In the open	3	9	34	5	...	51	3 661.6	6.8 } 22.5	
Total	3	25	143	11	...	182	4 023.9	22.5	...
Total general . . .	3	28	153	33	320	537	11 563.1	26.5	...

Number of train-miles: 81 491 140.
Total number of fractures: 537.

Number of fractures per 10 000 000 tr.-km. or 6 250 000 train-miles: 40.9.

* Standard gauge. — ** Most of these rails were put into service more than forty years ago

Characteristics of fractures of medium rails.

Fractures in the part of the rail:
— Covered by the fishplates: 150 = 82 4 %.
— Clear of the fishplates: 32 = 17 6 %.

— New and clean breaks through the whole of the rail section: 80 = 44.0 %.
— Fractures with old part: 102 = 56 %.

Rails broken into 2 pieces: 145 = 79.7 %.
Rails broken into 3 pieces: 24 = 13.2 %.
Rails broken into 4 pieces: 7 = 3.8 %
Rails broken into 5 pieces: 2 = 1.1 %
Rails broken into 6 pieces: 4 = 2.2 %

New and clean breaks through the whole of the rail section		Fractures with old part extending to the outer surface:		Fractures with much rusted old portion not extending to the outer face of the foot or head of the rail.
with oval mark.	without oval mark.	of the foot.	of the head.	
15 = 8.3 %	65 = 35.7 %	17 = 9.3 %	55 = 30.2 %	30 = 16 5 %

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
HOLLAND. Netherlands Railways.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	1	196.4	3	5	238.0	13	14	336 2	26	25	513.3	30	66	972.7	42	...	...	...	15 7
Medium.	2	252.3	5	6	227.4	16	25	256.0	61	8	296.4	17	2	27.3	45	...	...	...	19.7
Total . . .	3	418.7	4	11	565.4	15	39	592 2	41	33	809.7	25	68	1 000	42	...	...	...	

Number of train-miles: 34 176 030.

Number of fractures { total: 154. per 10 000 000 tr.-km. or 6 250 000 train-miles: 28.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . . .	64 %	36 %	No data.				
Medium rails . .	79 %	21 %					

		PERCENTAGE: *Light rails.*	*Medium rails.*	*Heavy rails.*
E. *a)* New clean fractures	with internal transverse fissure	4	...	Not used.
	without internal transverse fissure	26	19	
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	36	23	
	in the head	7	16	
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	27	42	
d) Number of pieces rails are broken into	2 pieces	75	77	
	3 pieces	22	14	
	4 pieces	3	9	

Names of administrations and description of rails.	Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
COLONIES. State Railways in the Dutch Indies.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light*	...	84.1	...	1	269.4	...	...	306.9	...	4	538.3	4.4	18	1 630.2	6.8	23	2 828.9	5.1	Steam tr. : 11.8. Elect. tr. : 13.8.
B. Rails in tunnels. *Light*	...	...	...	...	0.6	...	...	0.6	...	...	0.6	...	...	2.5	...	...	4.3	...	
C. The whole of A and B. *Light.*	...	84.1	...	1	270.0	2.3	...	307.5	...	4	538.9	4.4	18	1 632.7	6.8	23	2 833.2	5.1	

Number of fractures { total : 23. per 10 000 000 tr.-km. or 6 250 000 train-miles : 9.2. per 1 billion tkm. or 612 000 000 English ton-miles : 14.6

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	Number of fractures: on straight lines or curves of > 800 m. (40 chains) radius	Number of fractures: on curves of ⩽ 800 m. (40 chains) radius — Lower rail.	Number of fractures: on curves of ⩽ 800 m. (40 chains) radius — Higher rail.	Number of fractures: on a rising or falling gradient ⩽ 10 mm. per m. (1 in 100)	Number of fractures: on a rising or falling gradient > 10 mm. per m. (1 in 100)
D *Light rails.*	17.4 %	82.6 %	11	3	9	13	10
	Miles of single track of each class.		2 396.5	436.7		1 741.1	1 092.1

		Light rails.
E. a) New clean fractures	with internal transverse fissure	3
	without internal transverse fissure	5
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	15
	in the head	8
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	5
d) Number of pieces rails are broken into	not completely severed	10
	2 pieces	13

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Dutch Indies Railways Company.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels.																			
Light rails.																			
51.8 lb. D/M	...	...	...	...	...	...	...	...	...	...	...	...	3	21.1	88	3	21.1	88	
51.8 lb. G/S	...	...	...	...	...	...	...	...	...	...	...	...	...	105.0	...	...	105.0	...	
56 lb.	...	...	...	...	...	...	...	...	...	...	...	...	1	64.6	10	1	64.6	10	
60.1 lb	...	...	...	...	...	...	...	26.1	...	...	...	...	...	...	...	...	26.1	...	
67.5 lb	...	...	...	...	...	...	...	...	...	...	...	...	...	117.4	...	...	117.4	...	12.8
75 lb.	...	...	...	...	...	...	...	...	...	...	42.9	...	...	...	...	...	42.9	...	
76.6 lb	...	42.3	...	...	37.3	...	...	74.0	...	...	...	...	...	...	...	...	153.6	...	
82.7 lb	...	...	...	...	...	...	...	...	...	...	11.8	...	...	...	...	...	11.8	...	
Total	...	42.3	...	...	37.3	...	...	100.1	...	...	54.7	...	4	308.1	8	4	542.5	5	

Number of train-miles: 2 739 857.

Number of English ton-miles: 106 384 100.

Number of fractures: total: 4; per 10 000 000 tr.-km. or 6 250 000 train-miles: 9; per 1 billion tkm. or 612 000 000 English ton-miles: 22.99.

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	NUMBER OF FRACTURES: on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	NUMBER OF FRACTURES: on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	NUMBER OF FRACTURES: on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	NUMBER OF FRACTURES: on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails.*	...	100	3	1	...	1	1
	Miles of single track of each class.		51.8 lb. D/M = 21.1 miles 56 lb. M/W. = 64.6 miles	...	...	...	...

	Light rails.
E. *a)* New clean fractures: with internal transverse fissure	...
a) New clean fractures: without internal transverse fissure	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head: in the foot	1
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head: in the web	3
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head: in the head	...

d) Number of pieces rails are broken into: The 56-lb. M/W. 1 rail was broken into two pieces. Of the three 51.8-lb. D/M rails, 2 had their head crushed at the outer side and 1 at the inner side.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS: Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
CZECHOSLOVAKIA. State Railways.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	3	553.3	3.4	13	469.4	17.2	31	325.7	59 1	87	823.2	65.7	688	4 936.6	86.6	822	7 108.2	71.9	...
Medium.	6	794.6	4 7	14	574.5	15.1	16	272.3	36.5	22	338 5	40.4	55	751.2	45.5	113	2 731.1	25.7	...
Total . . .	9	1 347.9	4.1	27	1 043.9	16.1	47	598.0	48 8	109	1 161.7	58.3	743	5 687.8	81.2	935	9 839.3	59.0	
B. Rails in tunnels. *Light.*	...	5.7	...	...	2.8	...	...	2.3	...	1	2.3	273.8	1	4.1	152.5	2	17.2	72.5	...
Medium.	2	6.5	191.2	1	2.8	221.3	...	0.7	...	...	1.0	...	1	2.8	220.3	4	13.8	180.0	...
Total . . .	2	12.2	101.6	1	5.6	112.0	...	3.0	...	1	3.3	189.3	2	6 9	180.3	6	31.0	120.4	
C. The whole of A and B. *Light.*	3	559.0	3.3	13	472.2	17.1	31	328.0	58.7	88	825.5	66 2	689	4 940.7	86.6	824	7 125.4	71.9	12 8 to 15.8
Medium.	8	801.1	6.2	15	577.3	16 1	16	273 0	36.4	22	339.5	40.3	56	754.0	46.1	117	2 744.9	26.5	16.8
Total . . .	11	1 360.1	5.0	28	1 049.5	16.6	47	601.0	48.6	110	1 165.0	58.7	745	5 694.7	81.3	941	9 870.3	59.2	

Number of train-miles: 70 830 140.
Number of English ton-miles: 17 323 935 100.

Number of fractures: total: 941; per 10 000 000 tr.-km. or 6 250 000 train-miles: 82.6; per 1 billion tkm. or 612 000 000 English ton-miles: 33.2.

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES: on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails*	56.07 %	43.93 %	437	247	140	598	226
Medium rails	51.28 %	48.72 %	73	28	16	107	10
		Total . . .	510	275	156	705	236
		Mileage of single track of each class.	6 283.3	3 587.0		6 848.4	3 021.9

E.		*Light rails.*	*Medium rails.*
a) New clean fractures	with internal transverse fissure	165	24
	without internal transverse fissure	188	21
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	168	25
	in the head	192	25
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	111	22

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
TURKEY. State Railways.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	1	431.8	1.44	2	477.2	2.6	...	88.9	...	...	124.3	...	13	681.0	11.86	16	1 765.4	5.63	16.7
A. Rails outside tunnels. *Medium*	...	...	...	...	...	...	...	14.3	...	...	...	...	...	...	...	...	14.3	...	...

	Percentage of fractures in the part covered by the fishplates	Percentage of fractures in the part clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius.	on curves of ⩽ 800 m. (40 chains) radius: Lower rails.	on curves of ⩽ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient ⩽ 10 mm. per m. (1 in 100)	on a rising or falling gradient > 10 mm. per m. (1 in 100)
D. *Light rails.* . .	...	100	11	...	5	16	...

	Light rails.
E. *a*) New clean fractures with internal transverse fissure	13
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head — in the foot	2
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head — in the head	1

MISCELLANEOUS INFORMATION.

[621. 335 (.43)]

1. — German high-speed electric locomotive.

(Electric Railway Traction, supplement to The Railway Gazette.)

In *The Railway Gazette* for July 28, 1933, an account was given of a trial run on the recently opened Munich-Augsburg-Stuttgart electrified section of the German State Railways, in which were recorded almost unparalleled speeds and rates of acceleration. Al-

Fig. 1. — 3 000-H.P. express electric locomotive, German State Railways.

Fig. 2. — Traction motor and driving axle assembly.

though embodying experience gained in the running of previous 1-Do-1 locomotives, with a maximum speed of 68.5 m. p. h., the locomotive which made the run was to a new design, and by the courtesy of the builders, the Allgemeine Elektricitäts-Gesellschaft (A. E. G.), of Berlin, we are now able to describe the machine in detail.

Designated to give an output of 2 820 H.P. at 48 m. p. h. on the hourly rating, the locomotives of Class E04 are of the 1-Co-1 type with individual axle drive. Some of the class have been given a gear ratio suitable for a maximum service speed of 68.5 m. p. h., but the remainder are geared for 81 m. p. h. On a short-time rating the output at the motor shaft was calculated as 2 930 H.P. at 52.5 m. p. h., but during the course of the tests referred to above, outputs in excess of 3 000 H.P. were attained. The maximum tractive effort is

39 650 lb., corresponding to a factor of adhesion of 3.34, each of the three driving axles carrying a load of 20 tons. In working order the locomotive scales only 89.5 tons, *i. e.*, 68.5 lb. per H.P., or 32.8 H.P. per ton on the short-time rating.

A striking feature in the appearance of the locomotive is the long driving wheelbase of 19 ft. 8 3/8 in., with a distance of 13 ft. 5 1/2 in. between the trailing and centre driving axles. However, this does not affect the flexibility on curves, for the locomotive has no rigid wheelbase, the outer driving axles being combined with the adjacent carrying axles in Krauss-Helmholz trucks, and allowed 15 mm. (5/8 inch) play on each side.

Previous A. E. G. electric locomotives for the Reichsbahn have been built up on modified bar frames of lattice form, 50 mm. (1 31/32 inches) thick, but with a view to reducing the weight and cost of manufacture, plate frames 30 mm (1 3/16 inches) thick have been used for Class E04. Welding has been adopted for a number of details, including some of the frame stretchers, and the cab framework, which is shown mounted on the main frame in the accompanying illustration. The Kunze-Knorr air brake applies blocks to the driving

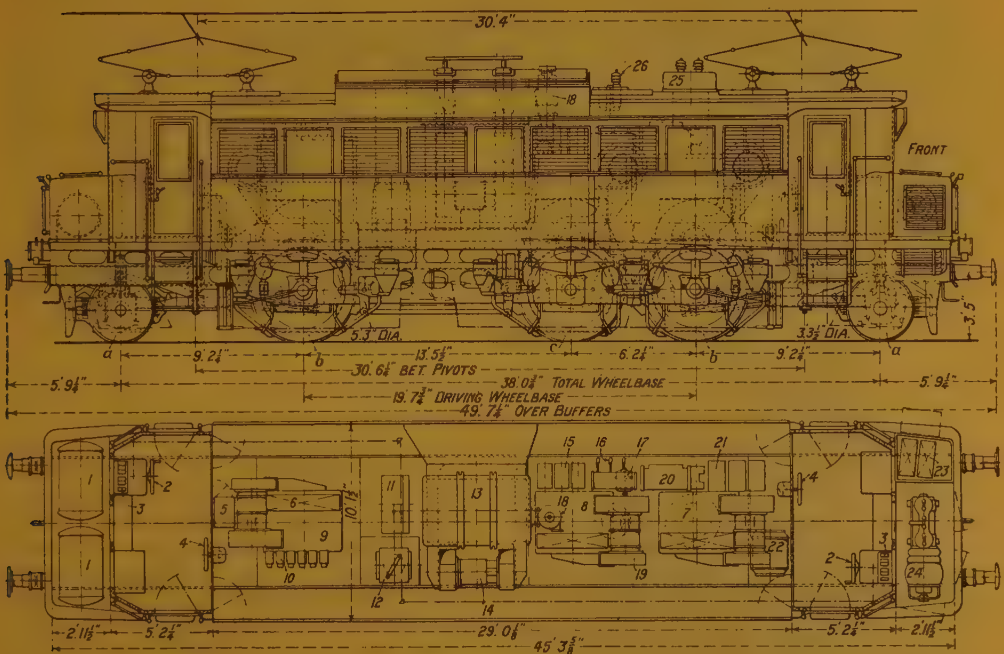

Fig. 3. — General arrangement of 3 000-H.P. electric locomotive, German State Railways.

1. Air reservoir.
2. Controller.
3. Meters.
4. Handbrake.
5. Locker.
6. Commutator cover.
7. Traction motor, No. 1.
8. Traction motor, No. 2.
9. Traction motor, No. 3.
10. Auxiliary switches.
11. Contactors.
12. Fine-grade controller.
13. Transformer.
14. Transformer blower.
15. Heater relays.
16. Motor isolation switches.
17. Lighting generator.
18. Oil pump.
19. Motor blower.
20. Reverser.
21. Sectionalising switches.
22. Auxiliary transformer.
23. Lighting battery.
24. Air compressor.
25. Circuit breaker.
26. H. T. insulator.

a ± 75 mm. play.
b ± 15 mm. sideplay.
c 10 mm. tyre thinning.

wheels with a force equal to 76 % of the adhesion weight, and on the locomotives arranged for a top speed of 81 m. p. h. the carrying wheels are also braked.

Each traction motor is mounted above its driving axle, and is bolted to the locomotive frame by brackets which transmit the torque reaction. The motor is also supported by two bearings on the hollow quill which surrounds the axle, and by means of twin gears drives the wheels through spring cups built into the end discs of the quill, as illustrated herewith. The Westinghouse drive has already been thoroughly tried out on the previous 1-Do-1 locomotives of the Reichsbahn, and on many other railways. The gear ratio is 3.415 to 1 in the locomotives with a top speed of 68.5 m. p. h., and 2.94 to 1 in the high-speed units. The pinions are of nickel-chrome steel with an ultimate strength of 83-89 tons per sq. inch, and the gear wheels are of similar material with a strength of 51-57 tons per sq. inch. Each quill is built up by welding the two end discs to the central hollow cylinder, and the gears are housed in two-piece casings of light metal.

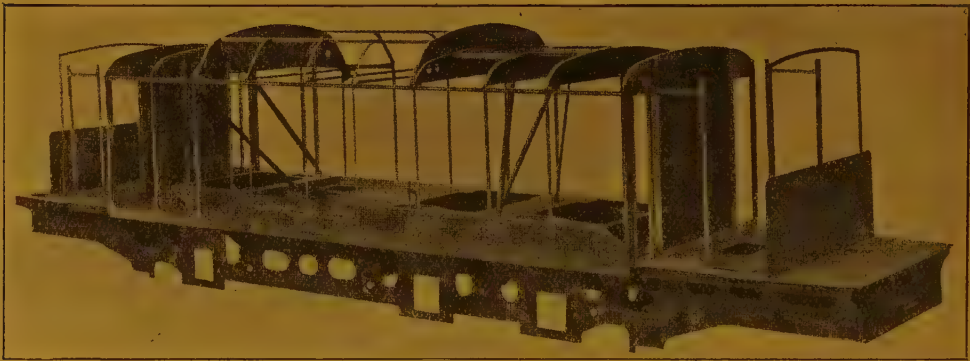

Fig. 4. — Main frame and cab structure.

In order to obtain a high rating per unit of weight, special attention was given to the cooling of the electrical apparatus. On the hourly rating 200 m³ (7 063 cu. feet) of air are passed through each traction motor per minute, and 170 m³ (6 000 cu. feet) through the transformer. The traction motors are of the single-phase series type with 12 poles, and run on ball bearings. On the continuous rating the oil-cooled transformer has a capacity of 1 400 *kVA.*, and has 15 tappings up to the maximum motor voltage of 550. The main control system is of the type used on many recent electric locomotives of the Reichsbahn.

[621. 135.1 & 621. 137]

2. — A clear engine cab outlook.

(From *The Railway Gazette.*)

The ordinary cab window of a locomotive is based on the principle that a solid transparent screen is the only means of enabling the driver to see ahead while protecting him from wind. Unfortunately, the glass which stops the wind also collects so much rain or snow, dust, soot, and grease that visibility is seriously impaired, often when clear vision is most required. At least a partial solution to the problem is provided by the

use of oscillating wipers where the wind screens of motor cars are concerned, but this device is inadequate to the requirements of locomotive service, partly for mechanical reasons and partly owing to the special difficulties associated with greasy soot, snow, and glazed frost. Where glass windows are used, the driver is compelled to expose himself frequently to wind, rain, snow and cinders, with detriment to his eyes and throat and risk of fatal injury. In 1930, there were 31 deaths on French railways alone to drivers or firemen being struck by obstacles alongside the track. Even if the driver opens the window or puts his head outside the cab he is apt to be more or less completely blinded by wind, rain, or snow at a time when the range of visibility of signals may be seriously reduced by climatic conditions; and to these objections there is added the confusing reflection of lights in cab windows when running through large stations or thickly populated districts.

(Editorial, *The Railway Gazette.*)

* * *

The advantages of direct vision at all times are secured in the Pottier cab outlook arrangement by the use of automatic aerodynamic screening instead of glass windows. As far as the driver is concerned, the outlook is through a plain opening, which, nevertheless, admits neither draught, rain, snow, nor cinders. Attempts to obtain this result by a blast of air across the opening have failed because such a screen cannot be kept effective at all speeds and under all conditions. In the Pottier arrangement, however, as developed by the « Société l'Aérodynamique Industrielle », the column of air tending to enter the opening is opposed by another column of air acting in the opposite direction. This opposing flow is derived from the incident air by means of deflector plates, and other plates are arranged to maintain reduced pressure in an evacuation channel which removes the whole of the air concerned, both incident and deflecting. Balanced air pressure is thus maintained outside and inside the opening, behind which the driver is therefore protected as completely as if glass were used.

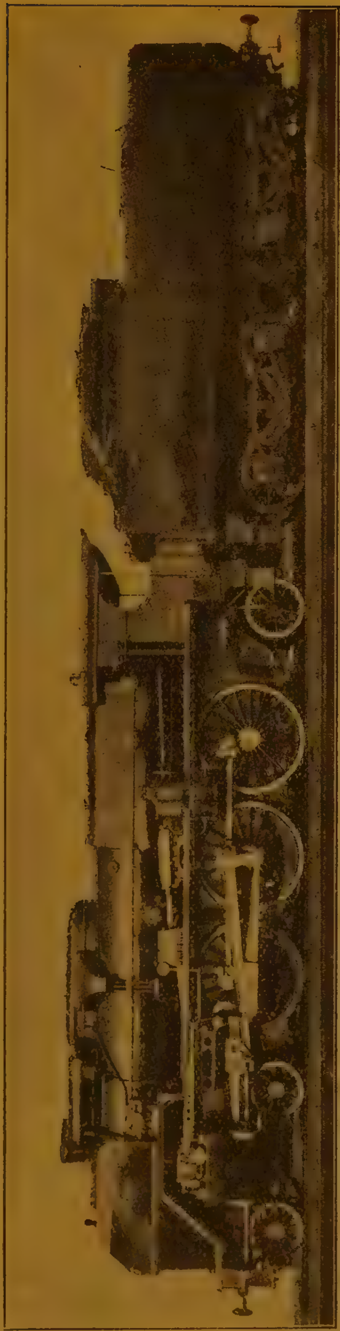

Fig. 1. — Nord 4-6-2 express locomotive rebuilt as 2-cylinder simple with Cossart valves and fitted with Pottier cab outlook. (See *Railway Gazette,* 17-3-33, p. 283.)

The form and arrangement of the deflector plates are necessarily different for each type of locomotive, but the principle is the same in all cases. The simplest possible arrangement is shown diagrammatically in figure 2. The direct flow of air *a*, which would ordinarily enter the opening *2*, is balanced and deflected by the current *b*, and withdrawn *via* the passage *c* behind the plate *5*. The flow of air over the outside of *5* effects a reduction of pressure at the rear of *c*, which acts in conjunction with the positive pressure in front of *4* to maintain the indicated conditions of flow. In this simple form, the apparatus requires the use of relatively large deflecting surfaces. The addition of supplementary deflecting surfaces as shown at *8*, *9*, figure 3, diminishes the kinetic energy of the direct column of air, *a*, to be deflected; at the same time *9*, in conjunction with *4*, amplifies the effect of the opposing and deflecting column *b*. Finally, the plate *8* enables the cross section of the discharge passage to be increased while still maintaining the same value of depression. As thus arranged, the apparatus has a large margin of safety, ensuring its satisfactory operation under all conditions of direction and intensity of wind. There are no moving parts and the apparatus requires no attention or maintenance.

Figures 1, 6 and 7 show the equipment installed on a locomotive of the Nord Railway of France. Following upon successful trials of the new outlook arrangement on a number of locomotives of the Nord Railway, that Company, the State and the Paris-Orleans Railways have placed important orders for the equipment of further locomotives, and deflectors are under design for a Paris-Lyons-Mediterranea *Pacific*. The drivers who have had experience with the apparatus speak highly of its performance under the most severe conditions, including running through a very heavy storm without any reduction of speed. Its universal adoption has been urged in connection with the Lagny disaster.

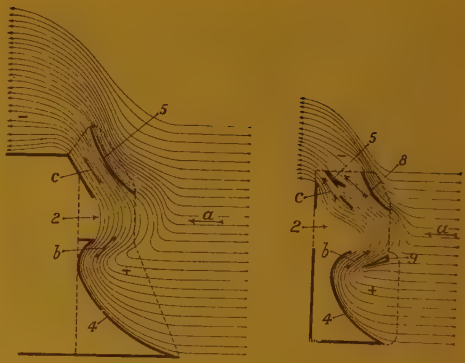

Fig. 2 (left). — The principle of the Pottier direct-vision cab outlook with aerodynamic screening; Fig. 3 (right). — Improved arrangement of deflectors resulting in more powerful screening.

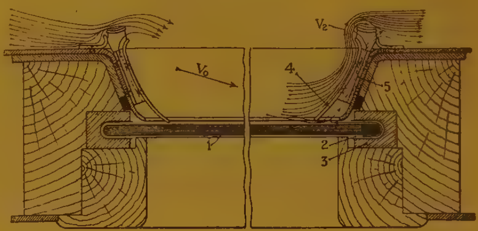

Fig. 4. — Carriage window protected against draughts and driven by aerodynamic deflection.

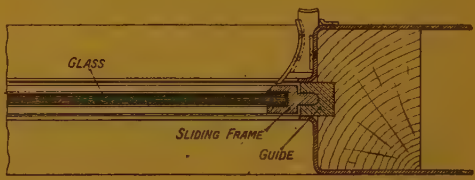

Fig. 5. — Aerodynamic deflectors applied to the window of an existing sleeping car.

Draught exclusion.

Precisely the same principles of aerodynamic deflection and pressure balance have been applied by Mr. Pottier to prevent the driving of air and water through the clearance spaces round the edges of carriage windows. His patents on this subject, as worked by the « Société l'Aérodynamique Industrielle », are illustrated by figure 4 in relation to a window designed spe-

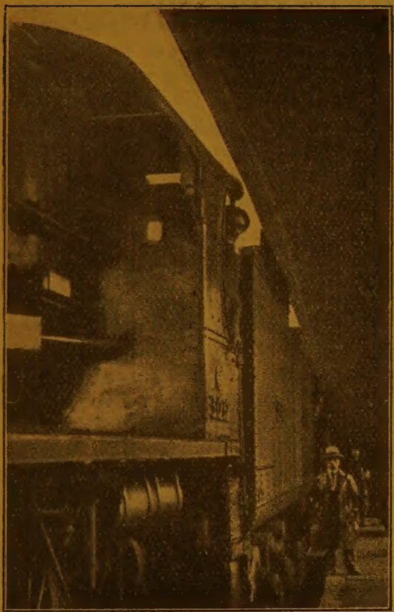

Fig. 6. — Nord *Pacific* locomotive with Pottier cab outlook.

Fig. 7. — Pottier cab outlook arrangement on Nord *Pacific*.

cially for this method of protection, and by figure 5 as regards the adaptation of the system to the window of an existing sleeping car. The essential feature is the diversion of the incident air in such a way as to establish zones of reduced air pressure drawing air, dust and water away from the clearance spaces through which they would otherwise be driven.

Referring to figure 4, the incident air, having a velocity V_0 relative to the window, would ordinarily be driven through the clearance *2*, between the glass *1* and frame *3*, into the interior of the vehicle. Actually, however, the portion of the air deflected by the plate *4* flows over its outer edge with a velocity V_2 which effects a reduction of pressure between *4* and *5* resulting in an outward flow, along this passage, of air entering at the inner edge of *4*. At the same time, there is actually a small outward flow of air *via 2* instead of a pronounced inward draught.

The precise form of the deflector and suction plates may be varied as required to suit the design of different windows. Trials conducted on coaches belonging to various French railways, among them the Est, State and Nord Railways, have given very satisfactory results.

NEW BOOKS AND PUBLICATIONS.

[656 .2]

WILLIAMSON (J. W.). — B. Sc., D. L. — **A British Railway Behind the Scenes.** *A Study in the Science of Industry.* — One volume (8 3/4 × 5 3/4 inches) of 210 pages, illustrated. — 1933, London, E. C. 4, Ernest Benn Limited, Publishers, 154 Fleet Street. (Price : 5 sh. net.)

The author's idea has not been to write a technical treatise on the operation of a railway system, but to give the general public in such a form and in such terms as would be understood by all, a broad view of the organisation and arrangement of the many units which constitute in some way, as the title indicates, the hidden or unperceived side of a railway. The railway selected is the London Midland & Scottish Railway, the largest British Railway Company, and a Company which also operates transport by road and by sea, and possesses in addition one of the largest hotel undertakings in Europe.

The book reviews in turn the different departments of the railway and stresses in particular the use made of modern methods of organisation and especially the belt method as applied to the construction and repair of locomotives, carriages and wagons, the creosoting of sleepers, etc. The chapters devoted to the description of the locomotives, rolling stock, permanent way, and signalling are interesting, as are those dealing with the organisation of the passenger stations and shunting yards.

Other chapters deal with different questions relating to electrification, to road and water transport, and to the hotels organisation of the Company.

The author ends by describing the function of the accountants and research departments of more recent creation; the former brings to light the need for investigation into one or another class of expenditure which appears particularly high. The problem so set is sent to the research department which collects all available information on the subject, analyses the problem and initiates the experimental work to be done, introduces the results obtained into practice, and lays down the new methods of working, the financial advantages of which are subsequently verified by the accountants department.

While the work is primarily intended to expose to the layman, in a language that everyone can understand, the unknown sides of working a large railway system, the expert will find it most interesting to read owing to the constant reference made to the principles of rational organisation and scientific research which have dominated the reorganisation of a great British railway.

A. C.

[621 .43]

E. SANTIAGO PUERTAS, Chief Engineer, Madrid to Saragossa and Alicante Railway. — **Automotores para ferrocarriles con motor de combustion interna** (*Rail motor cars with internal combustion engines*). — One volume (9 1/2 × 6 1/2 inches) of 87 pages, with 23 figures. — September 1933, Madrid, Association General de transportes por via férrea, Calle del Prado, 26.

This work forms Number 15 of the publications issued by the « Asociacion General de transportes por via férrea » (General railway transport association). The author compares the steam and internal combustion engines for rail traction pur-

poses, and railway working with steam locomotives and with Diesel rail motors : he shows the great advantages of the last method for passenger services an secondary lines. An important chapter is devoted to the present position of rail cars with internal combustion engines in the different countries; this information is given in great detail, especially as regards Germany, Belgium and Holland, and includes particulars of the vehicles, the operating costs and the financial results obtained with the different types used. The work then considers, from an economic point of view, a service of Diesel rail motor cars comparatively with a steam service in a concrete case, and ends by analysing the technical considerations to be kept in mind when studying the working of a service by rail cars. The work is very well documented and gives an interesting and quite up to date general view of the question of the use of rail cars with internal combustion engines on secondary and suburban lines.

A. C.

[625 (.460)]

D. ANTONIO MENDOZA VILAR, Manager of the Office for rolling stock standardisation on the Spanish Railways. — **Unificacion del material ferroviario** (*Standardisation of railway rolling stock*). One volume (9 7/16 × 6 1/2 inches) of 65 pages, with 22 plates. — October 1933, Madrid, Asociación General de transportes por via férrea, Calle del Prado, 26.

This volume is No. 16 of the collection mentioned in the preceding review. After giving particulars of the work done by the Committees on standardisation of rolling stock in the United States, Germany, England, and France, the author describes the efforts of the Spanish Committee which started work in 1925.

The standardisation so far effected deals more especially with goods wagons : wheels and axle boxes, spring gear, draw and buffer gear, brake gear, etc., and also with the permanent way materials. The author describes the present organisation of the work : constitution of the management committee, the technical commissions, and the designs office; the plan of the investigations undertaken and the future lines to be followed as regards standardisation. In an appendix, the book deals with the organisation of the Central Rolling Stock Designing Office (O.C.E.M.) of the French railways, the work it has done and the results it has obtained.

This volume is, therefore, an interesting contribution to the literature dealing with the standardisation of railway rolling stock.

A. C.